PRODUCT ENGINEERING

GEORGE F. NORDENHOLT, *Editor*

Design Determines Production

BE IT AN ASHTRAY OR BATTLESHIP, the design of a product establishes definite limitations on the speed and economics of its production. The engineer at the design board determines not only dimensions and arrangements of parts and assemblies, but of greater importance, he makes mandatory the production methods, machinery and manpower needed to manufacture the product. Thereby he fixes, within limits, the cost and speed of its production.

Thus it is that today, with production machinery at a premium, with a shortage of skilled workers and a threatened scarcity of certain materials, the design engineer holds the key position in our production program. He creates both the designs of the things to be manufactured and the designs of the machinery to produce them, and dictates the materials to be used, type of production machinery required and the degree of skill needed in the manufacturing operations.

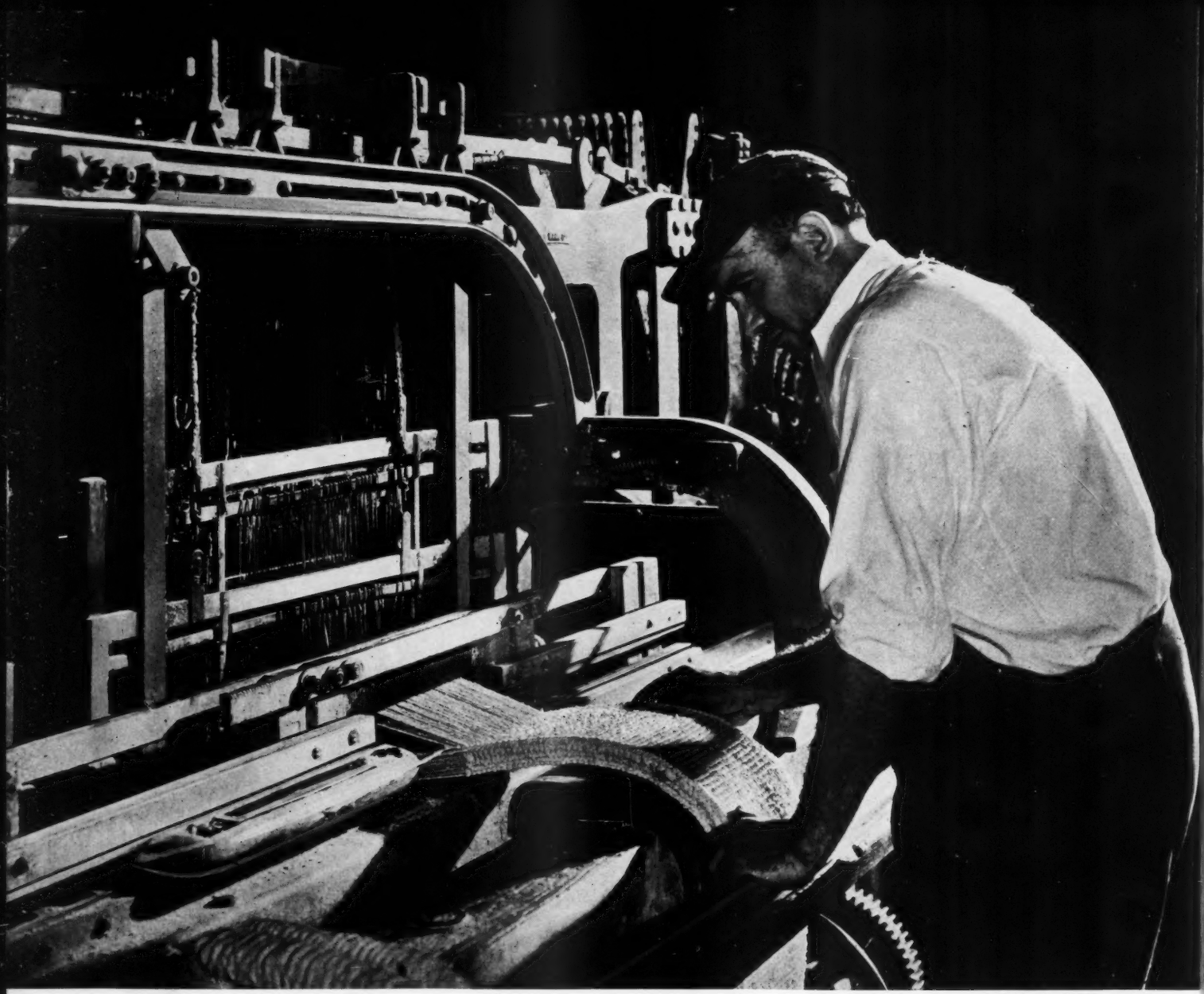
By ingenuity and the expenditure of much sweat, the designer is often able to increase appreciably the production speed, or to simplify the tooling required. By specifying larger dimensional tolerances he may eliminate certain finishing operations or perhaps make possible a less accurate but much faster production method.

By taking advantage of new developments in production technique, he might greatly reduce the amount of production equipment required or increase the production speed. The final decision as to whether the part shall be a casting, forging, stamping or molded piece will fix not only the cost and serviceability of the part, but will also determine the production machinery and skill required for its making.

It is becoming increasingly apparent that before many months have passed the dominating factor in the manufacturing industries will not be the dollar and cents cost of products, but the speed with which they can be produced with the materials, machinery and skill available. Cost is always a factor, but if the design calls for material that is unobtainable, machinery that is not on hand and cannot be bought, and skill that has not been trained, the design will be worthless.

Now as never before, prejudice and tradition must be thrown to the winds by the engineers and executives responsible for the design of machinery and other products. Thorough studies must be made to the end that the design will permit the part to be made with maximum speed with the production facilities and materials most readily usable and available.

Admittedly, engineering departments must be staffed with enough of the right kind of men. The executive engineers should have enough time that they can devote to investigating new design possibilities, to finding out about new design developments, to studying the technical literature and to making personal contacts with others in their profession. Such a program, applied systematically will be reflected in speedier production and earlier deliveries through design simplification.



Johns-Manville Corporation

Weaving thick asbestos lining on a giant loom, after which the lining is impregnated and molded to size by heat and pressure

FRICTION MATERIALS—I

Linings and Blocks for Brakes and Clutches

FUNDAMENTALLY any type of band, shoe, disk brake, or clutch is designed with the objective of retarding rotation in order to decrease or to stop the relative motion between brake or clutch members, by converting the kinetic energy of the moving system to heat energy.

A major consideration in the solution of braking or clutching problems is concerned with the variation of the coefficient of friction of the friction material during deceleration or acceleration. Factors which must be considered in determining the operating characteristics and capacity of a brake or clutch are the effects on friction values caused by temperature rise, maximum operating pressure, moisture, rubbing speed, wear, condition of brake drum surface, and length of time between operations.

No single friction material can meet the widely different conditions of brake temperatures, loads, drum materials and finishes, deceleration rates, and frequencies of application encountered in the various services. In the industrial field some services require flexible lining, others require dense rigid blocks.

The quantity of heat generated at the contacting surfaces of the brake lining and the drum for any given braking operation has a definite value dependent upon the difference in the kinetic energy in the moving system

before and after deceleration, regardless of the time taken to stop or decelerate the moving load. The brake factors on which converting the kinetic energy into heat energy depend are the coefficient of friction between the lining or block and the drum, the total normal contact pressure, and the rubbing speed.

Friction linings are good heat insulators, metal drums are good conductors and therefore readily absorb the heat produced by the frictional forces with an accompanying rise in temperature. Naturally the maximum temperature developed in the drum will depend upon the frequency of operation, the mass and type of material of which it is made, the exposed area of the drum and its ability to radiate heat.

The high resistance to heat possessed by friction materials made with an asbestos base, as compared with similar properties of wood, paper, fiber and leather, are the chief reasons for the preference shown asbestos friction materials in the automotive, industrial and construction machinery fields. The typical values of operating factors shown in Table I are for asbestos composition linings and blocks.

Effect of Temperature

Coefficients of friction for asbestos composition materials at temperatures above 500 deg. F. become erratic, rising to high values or dropping to low values with consequent lack of control of the rate of deceleration. Therefore, for satisfactory operation brake drums must have the ability to dissipate heat rapidly enough to prevent the temperature of the drum from becoming excessive. The temperatures at which variation in the friction value occurs are those at which the binder in the friction material undergoes a physical change of state, and varies greatly with different types of brake lining depending upon the heat resisting qualities of the binder or fillers after curing. The curves shown in Fig. 1 were established by dynamometer tests with various linings and indicate how with speed and pressure constant an increase of temperature may affect the coefficient of friction.

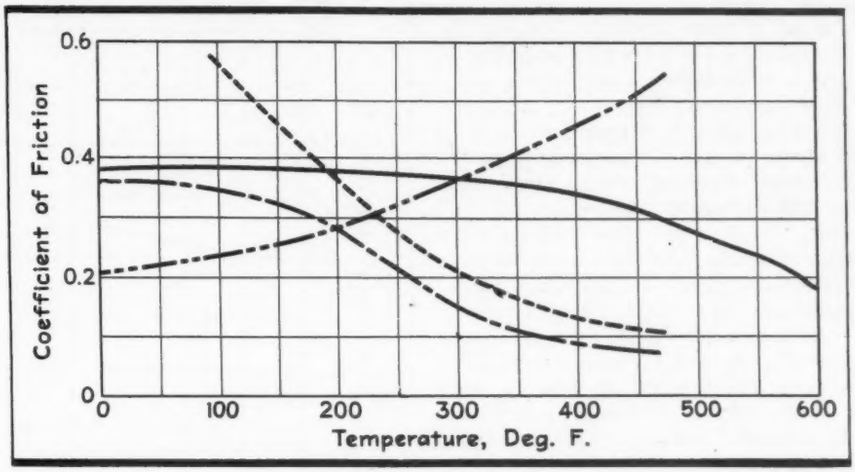


Fig. 1—Tests on various linings indicating how increasing temperature may affect frictional value under conditions of constant speed and pressure

Although the binding materials are heat-treated in manufacture, the temperature on the surface of the friction material in service is often higher than that employed in manufacturing, with the result that the impregnant or binder may soften or even approach liquification thus reducing the coefficient of friction to low values and then to increase it as the binding materials are driven off developing a possibility of running on dry asbestos with a friction value of approximately 0.8. If a carbonaceous condition develops, the friction value may be higher or lower than normal, depending upon the character of the surface formed.

For satisfactory operation and long life of lining, brake drum temperatures should never exceed 400 deg. F. for rubber bonded friction materials, nor 600 deg. F. for resinous materials. With the latter the higher temperature is permissible since the majority of the synthetic resins used in the manufacture of friction materials are thermosetting and are rendered infusible by the curing temperature.

High temperatures also break down the mechanical strength of asbestos fibers and destroy their structure. Asbestos occurs in nature in many different forms and chemical compositions. The form used in the manufacture of friction materials is a magnesium silicate containing about 14 per cent water of crystallization. At temperatures at or above 700 deg. F. some of the water of crystallization is driven off, accompanied by a decided loss in strength of the fiber, in addition there may be a loss of organic impregnant through destructive distillation. At temperatures of about 1,000 deg. F. the magnesium silicate is rapidly dehy-

Table I—Performance Characteristics of Asbestos Linings and Blocks

These data cover only the best types of materials when used in dry service conditions.

Style	Coefficient of Running Friction	Rubbing Speed of ft. per min. max.	Drum Temp. Max. Deg. F.		Pressure lb. per sq. in.	Resistance to Shock	Rate of Wear
			Short Periods	Constant			
Folded and compressed rubber compound	0.30—0.40	3,000	500	250	100	Excellent	Excellent
Flexible woven	0.35—0.45	5,000	500	250	50	Fair	Fair
Heavy-duty woven	0.35—0.50	7,500	750	500	100	Good	Good
Molded linings	0.30—0.40	5,000	750	500	100	Good	Good
Blocks	0.30—0.40	5,000	1,000	750	150	Excellent	Good

drated, resulting in a chalk-like substance without mechanical strength. Therefore, it can be readily understood why it is necessary to keep drum temperatures below 700 deg. F. in order to assure satisfactory operation.

Increasing the normal braking pressure will, of course, increase the decelerating force applied to the drum, however, this is accompanied with slight tendencies either to build up or to decrease the coefficient of friction in most materials. Hence for any application the increase in decelerating force is generally in direct proportion to the increase in applied pressure. The curve in Fig. 2 was established from dynamometer tests on samples of linings at constant speed and temperature, and indicates how an increase in unit contact pressure may influence the coefficient of friction. Examples of the effect of increasing normal braking pressure on deceleration rate are shown in Fig. 3 for various coefficients of friction. Dry, unlubricated friction is independent of the area in contact for constant conditions of velocity, temperature and surfaces. Increased area, however, is obviously highly beneficial in reducing wear.

At low speeds of a few feet per hr. most friction materials have a coefficient of friction which is less than their values at high operating speeds. When the rate of sliding approaches zero just before stopping, it is thought that the coefficient of friction rises sharply possibly because of a fade at higher speed and an antifade on cooling of the contact surfaces. A drop in frictional resistance as the velocity approaches zero is desirable. If the low speed frictional resistance is considerably below the normal friction value, however, the holding ability of the brake may become uncertain. Since if for any reason a slight movement or vibration of the drum were induced in a standing or suspended load, the frictional grip of the brake would immediately drop and the load might start and continue to creep indefinitely.

Some tests have shown that with increases of rubbing speeds up to about 500 ft. per min. there is a gradual increase in the coefficient of friction, and that at rubbing speeds between 500 to 5,000 ft. per min. there is a small dropping off in the friction value as the velocity increases, however, this reduction is small.

The rate of wear of asbestos friction materials varies considerably in different types. In order to maintain a friction level within dependable limits some wear is desirable to remove the smooth glazed surface which forms on the lining because of accumulations which take place. Oil or grease may

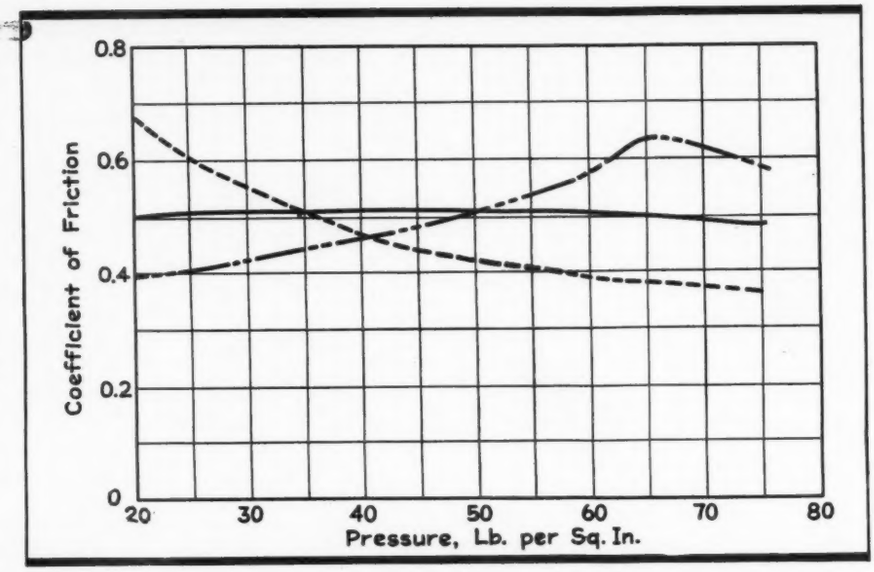


Fig. 2—Increase in unit contact pressure is accompanied with slight tendency either to build up or to decrease the coefficient of friction in most materials

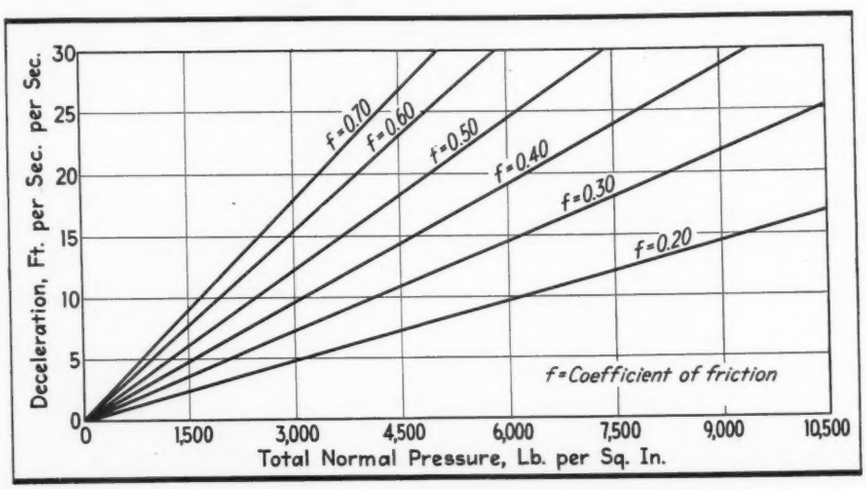
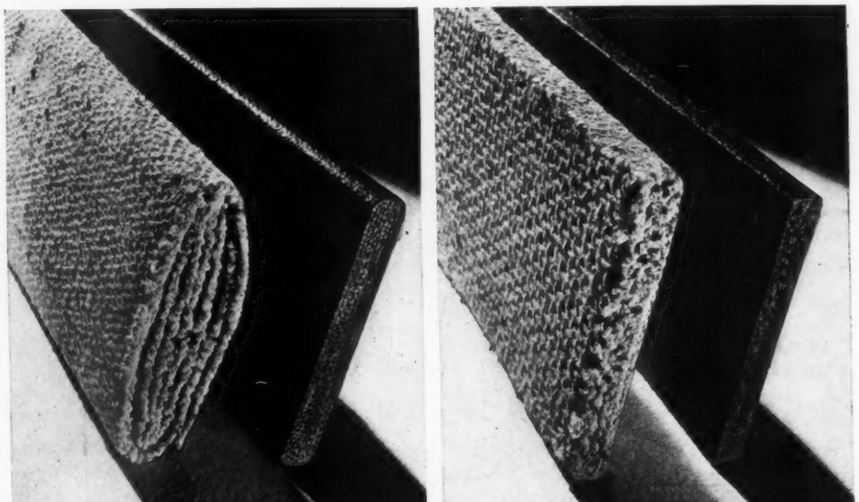


Fig. 3—Examples of the effect of increasing total normal braking pressure on the coefficient of friction



Johns-Manville Corporation

Fig. 4—Asbestos linings. Left, folded type. Right, flexible woven type

cause the lining to glaze. As glazing develops the coefficient of friction fades. Rough drum surfaces naturally increase wear and develop an apparent high friction value.

The effect of water on asbestos friction materials is to form a lubricating film, hence with some linings dangerous slipping occurs when the material is wet. This film during braking operations, however, is quickly squeezed out if not renewed. The rate of recovery in friction value depends upon the amount of water absorbed by the asbestos fibers. Asbestos is a hygroscopic material, it will absorb from two to four per cent moisture from the atmosphere. During manufacture the asbestos fibers are sometimes waterproofed as well as covered with the impregnating material. But as the friction lining or block wears in service, the fibers at the surface become uncovered and absorb water in addition to that held in the interstices of the lining structure and in the brake dust, which may when the lining has not worn uniformly account for the grabbing effects of asbestos friction materials when damp. The curves in Fig. 5, taken from Prony brake tests, indicate the effect of water on the frictional value of linings and their ability to recover from the lubricating effect of the water film. The ideal lining, of course, would be unaffected by water. Some linings, in recovering from the lubricating effect of the water film, become very sensitive and remain so for a considerable time. Others recover their prior effectiveness and remain at this level.

Some linings have a tendency to swell with the absorption of water so that they drag on the drum surface if the clearance is small. Oil may also cause linings, particularly rubber-bonded linings, to swell and soften. Some synthetic resin-bonded linings are not physically affected by a thorough soaking in oil; their frictional value, however, drops considerably. Oil on a rubber lining, on the other hand, usually increases the frictional value after the consequent softening and disintegrating of the rubber.

For satisfactory braking operations lining materials should be fitted tightly to bands with full bearing on the backing surface to prevent wrinkling and consequent crowding. All rivets or bolts used for fastening the lining should be body tight in the band in order to prevent any "play" that might cause a rivet failure by shear if subjected to shock or impact.

There should be no clearance on the bolts in line with the force on the blocks, as the blocks may slip and shear the bolts. All bolts and screws should have heavy steel lock washers to prevent loosening under vibration.

For flexible industrial linings larger than ½ in. thick brass bolts are generally employed because the shearing effect increases with the difference in radius of the outer and inner lining surface. Flexible linings less than ½ in. thick are usually secured to the bands by rivets. In Fig. 6 are shown various methods for fastening.

Fastenings should never be permitted to come in contact with the brake drum. The heads should be countersunk below the friction material surface as far as can conveniently be done without weakening the holding power of the fastening. In flexible materials, it is customary to counterbore at least half the thickness; in rigid materials about two-thirds the thickness.

Because of wear of the friction material, it is sometimes impossible to prevent occasional contact of the fastenings with the drums. Every effort should be made, therefore, to prevent the possible cutting, scoring, or wearing away of the drum by the proper selection of the fastening material. Rivets and bolts for fastening friction material to bands are usually made of brass, copper, or aluminum.

Brass rivets are superior to copper because they abrade rather than drag or plate the drum surface and show a minimum tendency to score under normal operating conditions.

As a general rule, solid flat head

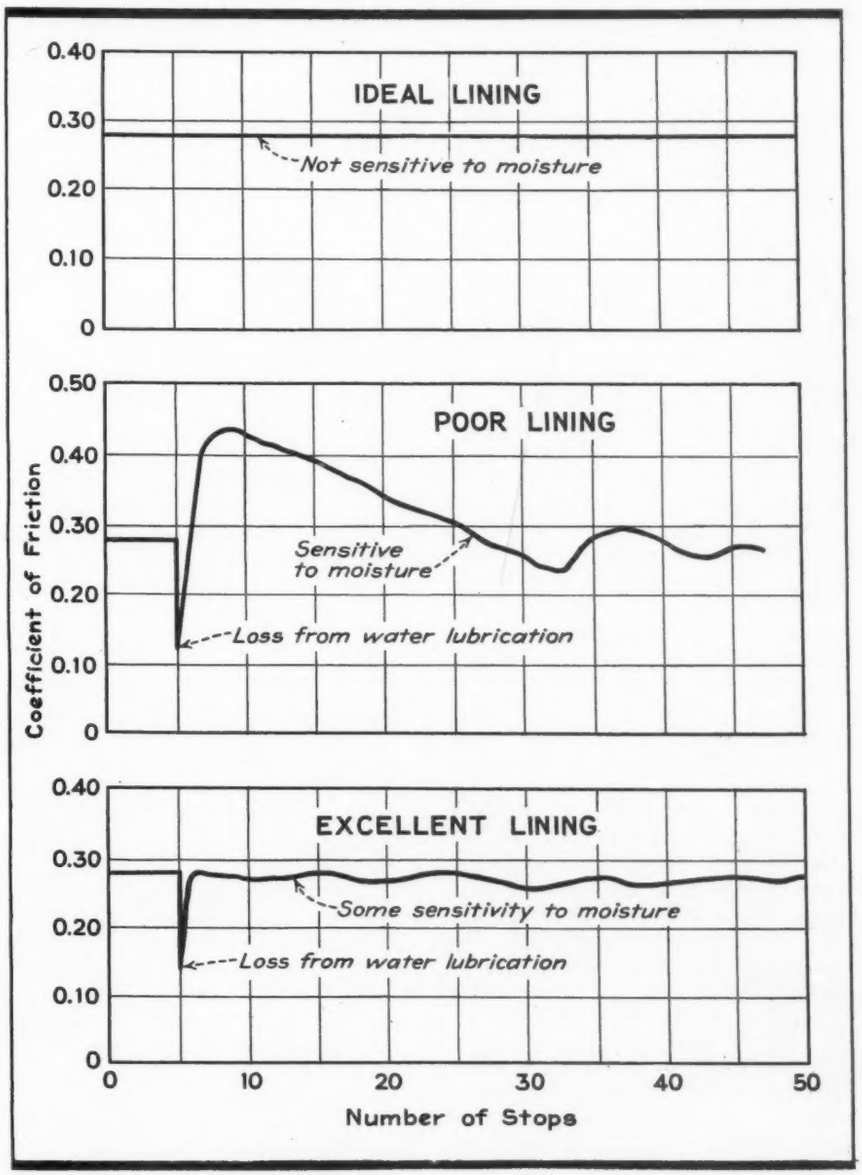


Fig. 5—Curves from Prony brake tests indicating the effect of water on the frictional value of linings and their ability to recover

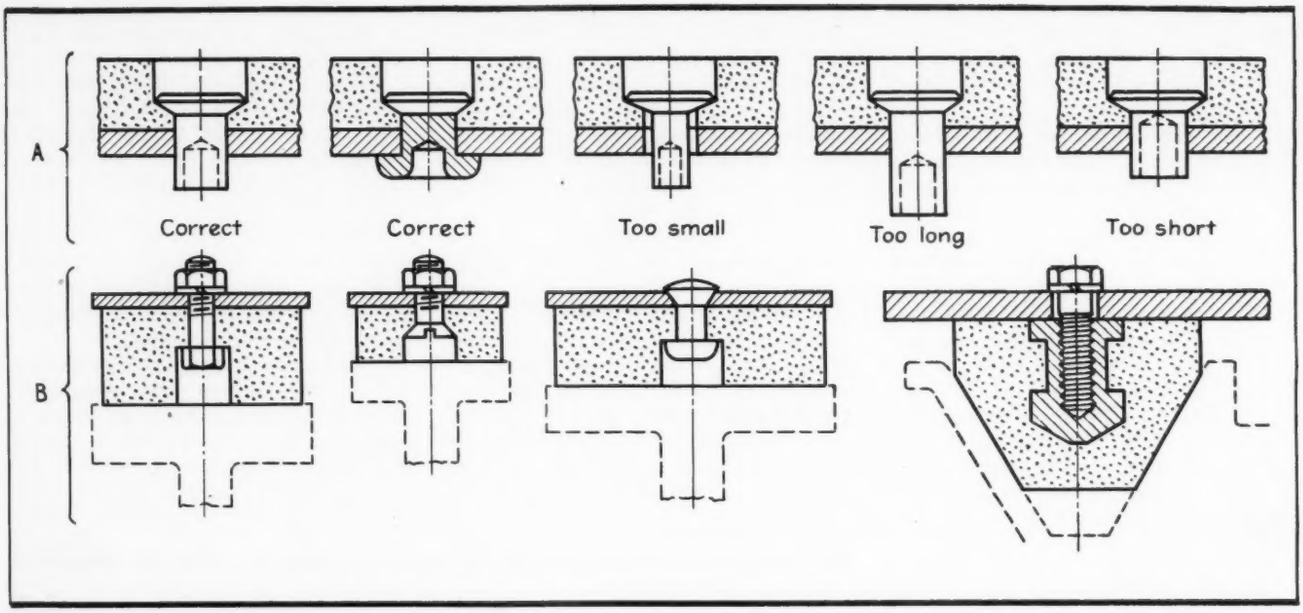


Fig. 6—Methods of fastening friction materials to brake bands. (A) Correct and incorrect methods for riveting linings to brake bands. (B) Methods of fastening blocks to bands

rivets are the most satisfactory type for industrial purposes. Semi-tubular rivets are better for this work than the full tubular kind as they can be spread more easily when upsetting to make a tighter grip. Split rivets should not be used on industrial equipment under any considerations.

The minimum length of rivet should be the thickness of lining under the rivet head, plus the thickness of the brake band or shoe, plus the diameter of the rivet body. The maximum length should be not more than one and a quarter times the minimum length.

For fabric linings it is suggested that rivet heads be made twice the diameter of the body or shank. Molded linings are dense and mechanically strong and do not require rivets with as large heads to hold the lining in position.

The number of rivets to use in a lining will depend upon the maximum pressure of engagement, the maximum coefficient of friction, and the safe shear strength of the rivets. The area A of brake lining in sq. in. per rivet, may be expressed in terms of maximum pressure of engagement p in lb. per sq. in., maximum coefficient of friction f, and safe shear strength S in lb. per rivet, as

$$A = S/pf$$

Safe shear strength of brass rivets based on a unit strength in shear of 12,000 lb. per sq. in. can be listed as

5/16 in. dia. rivet.....920 lb.
¼ in. dia. rivet.....600 lb.
3/16 in. dia. rivet.....300 lb
⅛ in. dia. rivet.....150 lb.

Rigid friction materials ⅜ in. thick and smaller are termed molded linings; and thicker pieces are known as friction blocks. Because of their high mechanical strength, rigid molded friction materials are often used where high stresses would break down the softer and more flexible types of materials. Rigid molded materials being formed under pressure and the binder set to a rigid and infusible condition usually are more resistant to heat and more constant in frictional values over a relatively wide range of temperature than are the flexible types. Rigid molded linings are also less compressible and so require less frequent adjustment when high pressures are employed.

Molded linings in roll form are usually made with an integral backing material to increase their mechanical strength, and are suitable for either internal expanding or external contracting wrap band or shoe type brakes. These linings have the friction facing on one side only and have to be curved to suit design with the backing on the proper surface, that is, internally for expanding and externally for contracting brakes. Coefficients of friction range from 0.20 to 0.50. The linings are highly resistant to wear and are suitable for rubbing speeds up to 5,000 ft. per min., drum temperatures up to 500 deg. F. constant service, and up to 100 lb. per sq. in. brake pressure. Flexibility depends on incompletely curing the organic binder and this type is more inclined to fade at high temperatures.

In recent years manufacturers are trying to design brake linings that have definite friction values and to include a graded range of these values in their line. It has been found, however, that some brakes require an overall frictional value, where there are two or more shoes, and this value is not satisfied by any one lining. Some brake designers use different types or styles of lining for different brakes in order to select the friction and wear characteristics that are most satisfactory.

To provide in-between friction values, two different linings may be used on a given brake. One type of lining may be used on an energizing shoe and another kind on an auxiliary shoe. In the two-shoe type of brake, the energizing shoe does more of the braking than the non-energizing shoe and wears faster. To compensate for this increased wear, one automobile manufacturer furnished a thicker piece of lining on the energizing shoe. Another type of two-shoe brake has a longer piece of lining on the energizing shoe than on the non-energizing shoe. Some lining segments are so tapered that they are thicker at the areas of more rapid wear yet fit the drum.

EDITOR'S NOTE: For contributing data and helpful criticism during the preparation of this article, the editors acknowledge their appreciation of the valuable cooperation of J. W. Hemphill, Jr., Johns-Manville Sales Corporation; T. R. Stenberg, Marshall-Eclipse Div., Bendix Aviation Corporation; and Dr. F. C. Stanley, Raybestos Div., Raybestos-Manhattan, Incorporated.

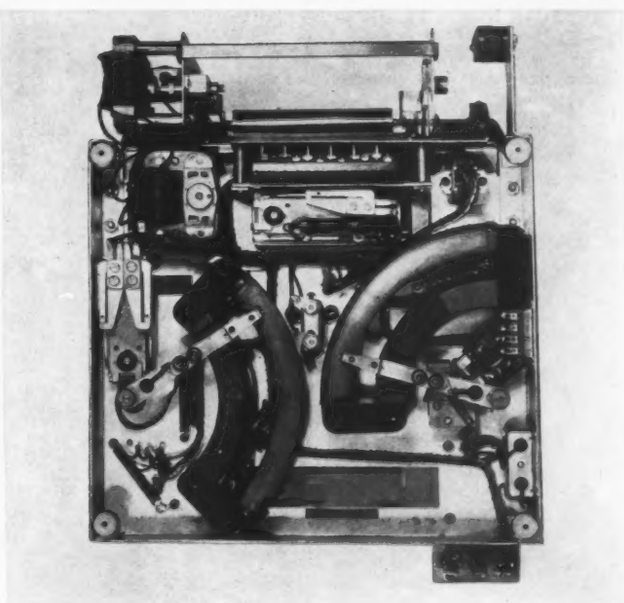
Modern Designs

Redesign Reduces Costs

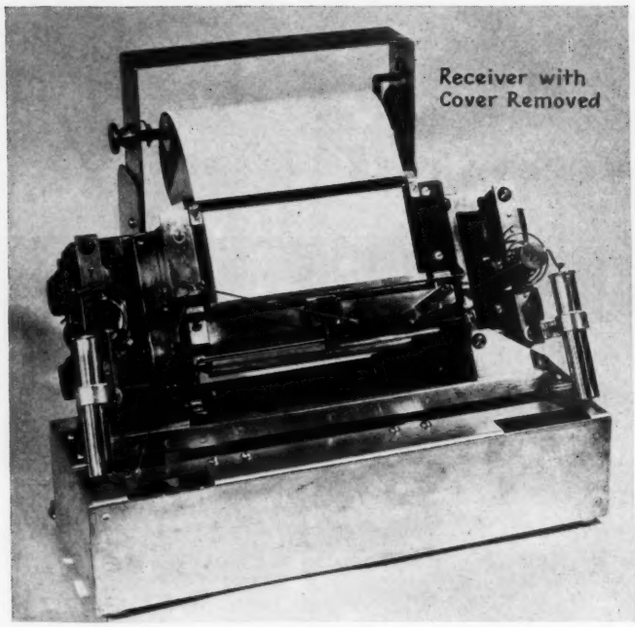
Production economies are obtained through simple forms, inexpensive dies and ease of assembly in redesign of the TelAutograph transmitting and receiving sets. After analyzing the functions of the machines and studying the engineering and production considerations, Vahan Hagopian, industrial designer, with the cooperation of TelAutograph engineers, designed new models constructed of simple parts, with pleasing lines and good proportions.

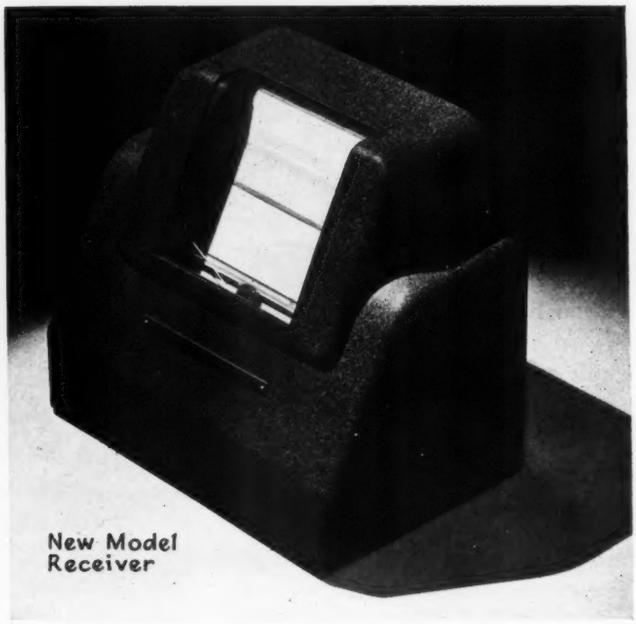
Two signals are sent out simultaneously by the transmitter, proportioned and controlled by the plane coordinates of the motion of the writing stylus. Transmitter signals are translated in the receiving set into similar coordinate positions of a writing pen. A third simultaneous signal controls lifting and depressing motions of the pen. Pressure of the stylus on the writing surface controls the signal that depresses the receiving pen onto the paper.

(Continued on next page)



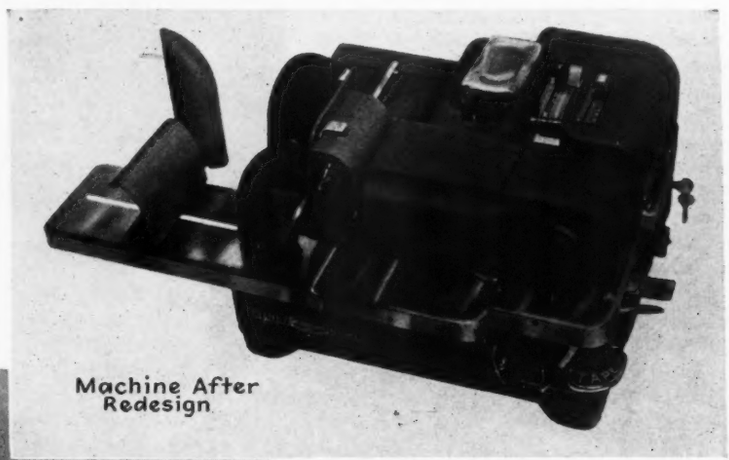
MODERN DESIGNS – Redesign Reduces Costs *(continued)*

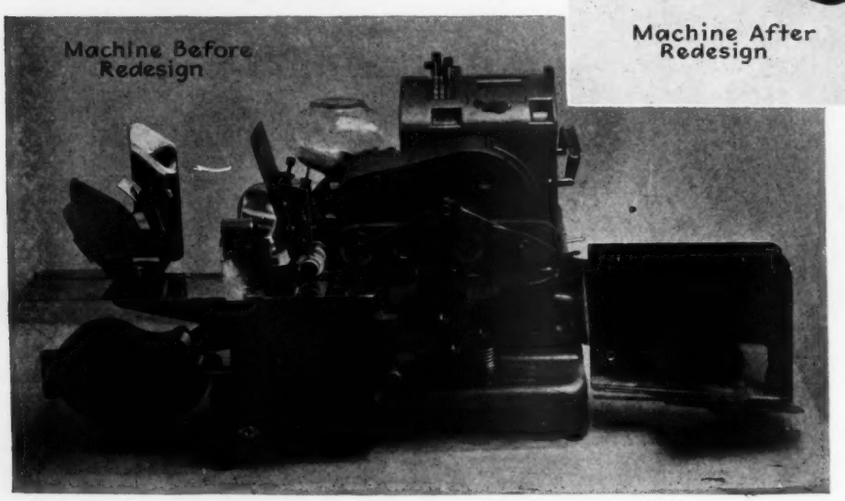




Formed and welded sheet metal parts are used for chassis, casing and cover because these machines are not made in large quantities, thus avoiding the expense of complicated dies. Casings are made of sheet steel. Welded steel chassis supports the casing and also the mechanism. Writing plate is a sheet of Monel metal. Casings have a wrinkle finish so that incidental surface scratches will not be conspicuous. Parts of casing that the operator must remove by lifting when changing the paper rolls are on reassembly, aligned and supported against rabets.

Machine bases, formerly made of cast iron, are now stamped from No. 14 gage sheet steel to form an inverted pan with flanged edges and with diagonal ribbings to obtain strength and stiffness, in the new Pitney-Bowes postage meter machine, redesigned by Vahan Hagopian in cooperation with the development engineers of the Pitney-Bowes Postage Meter Company. Zinc die-castings are used for most of the parts comprising the casing.



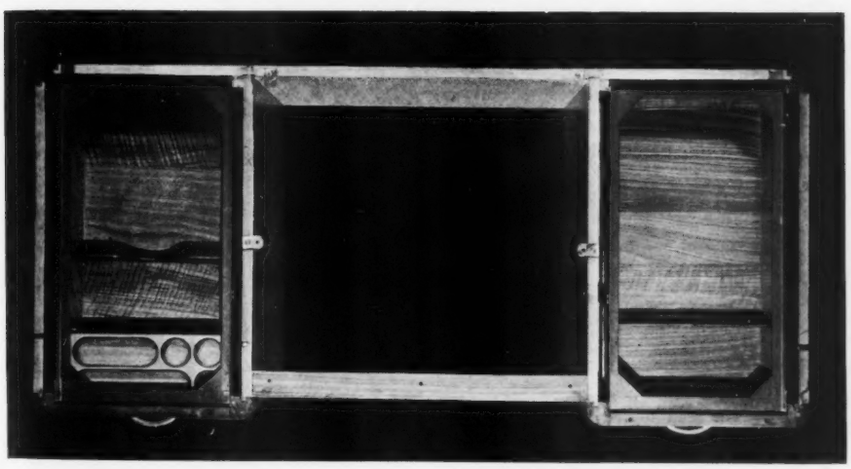


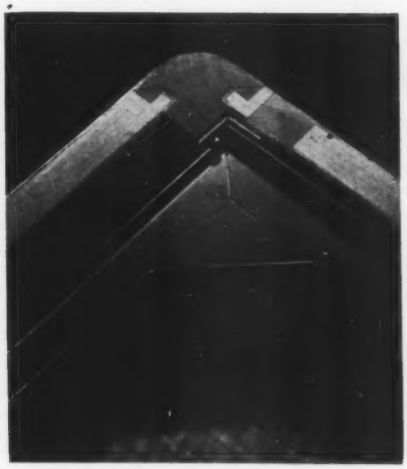
Carrying handle of the machine is a stainless steel stamping. Other handles and knobs are zinc die-castings, these are grouped in convenient positions facing the operator. To reduce the number of dies needed many parts were designed so that they can be used in different machines. Stainless steel because of its strength and resistance to wear was selected as the material from which to stamp the feed plate. The edges of the feed plate are skirted to gain strength. Housings are coated with a wrinkle finish.

Interchangeable Desk Units

Developed by the office management division of the Studebaker Corporation and engineers of the Stow-Davis Furniture Company, this "convertible" desk is designed so that eleven different types of desks can be made from one base, a typewriter platform, and various sizes of tops. Construction is laminated wood and welded steel frames which contribute greatly to strength and durability. "Sled runner" bases are solid stock and hold the bottom of the desk 7 in. above the floor. To the right is shown the desk with "executive top" which overhangs 9½ in. at each end and 9⅜ in. at the back. Top is 5-ply, 1½ in. thick American black walnut, banded on four edges with solid stock tongued to core and molded flush with top.

Clerical desk is equipped with a typewriter shelf 19 in. by 16½ in., which can be placed at the right or left-hand end of the desk. On all styles and models the drawer pulls are aluminum alloy die castings, with bronze plated finish. Bases are banded at the bottom with a protective metal strip finished to match the drawer pulls. All drawers have 5-ply flush front, full dovetailed construction with 3-ply bottom, and are completely interchangeable. Finish is nitro-cellulose lacquer, highly resistant to alcohol and other liquids, hot or cold. Standard colors are brown walnut or silver gray walnut, hand rubbed to a smooth dull gloss. All pedestal bodies are 58 in. long, 26⅜ in. wide.

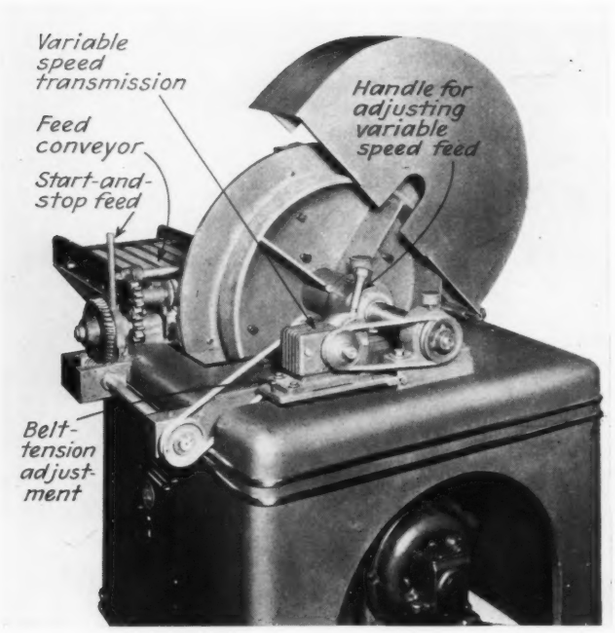




Top view of pedestal with top removed, showing some of the welded steel framing, stiffened by corner gussets. Metal partition clips, bronze finished, are adjustable to hold the partitions at any position. The tops are attached to the pedestal with machine screws which pass through the steel clips fastened to the top edge of the pedestal and are threaded into steel inserts which are embedded in the underside of the plywood desk top.

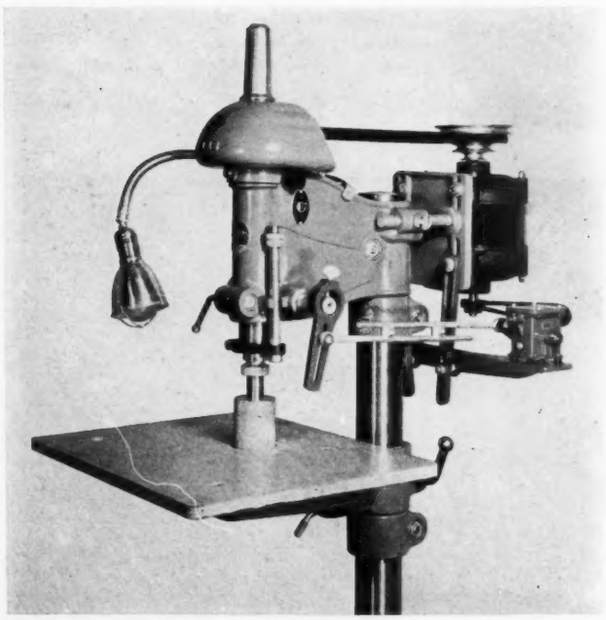
MODERN DESIGNS – Welded Frame, Adjustable Feed

Paper shredder built by Mitts & Merrill has rotating knives which cut newspapers, magazines and waste paper into narrow strips that are fluffed to make "paper excelsior" used for packing. Speed of feed determines width of cut strips or shreds. Base of machine is welded steel plate.

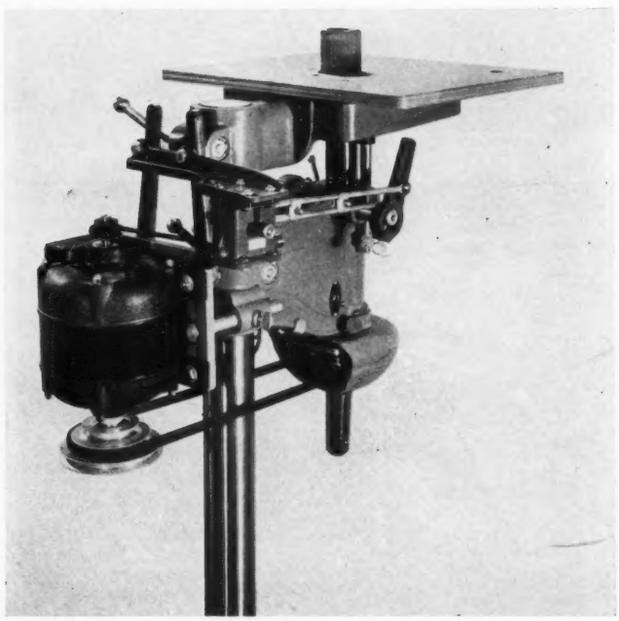


Speed of feed conveyor and feed rolls is adjustable by shifting handle of the Select-O-Speed transmission unit. Tension of V-belt drive to countershaft is adjusted by raising or lowering hinged base of the transmission unit. Flame-cut opening in welded base gives convenient access to motor.

Unit Converts Drill Press into Drum Sander

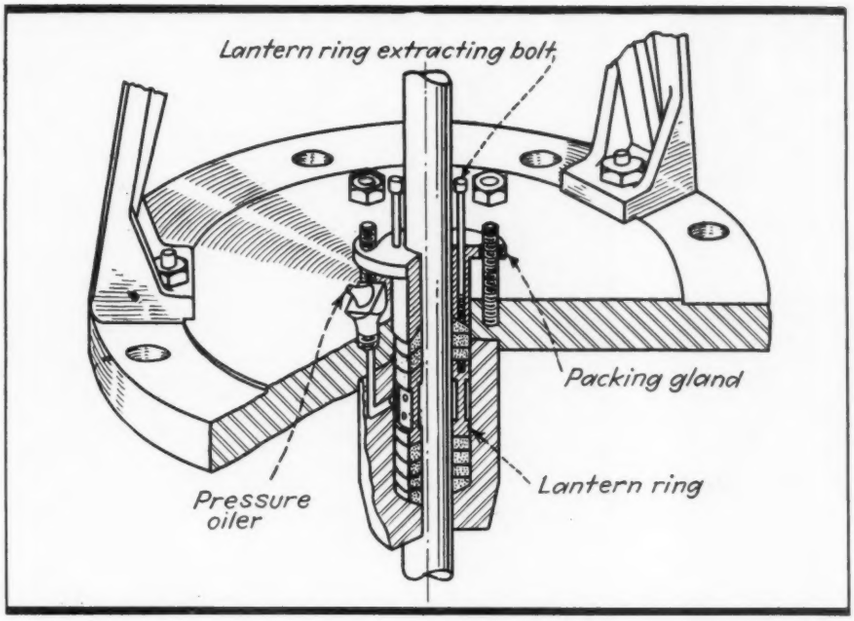


Reciprocating drum sanders can be made from standard drill presses through the addition of a unit, Reciproc adjustable oscillating mechanism, built by Specialty Machinery Company, to reciprocate drill press spindle. Conventional mounting on a three-speed drill press is shown above.

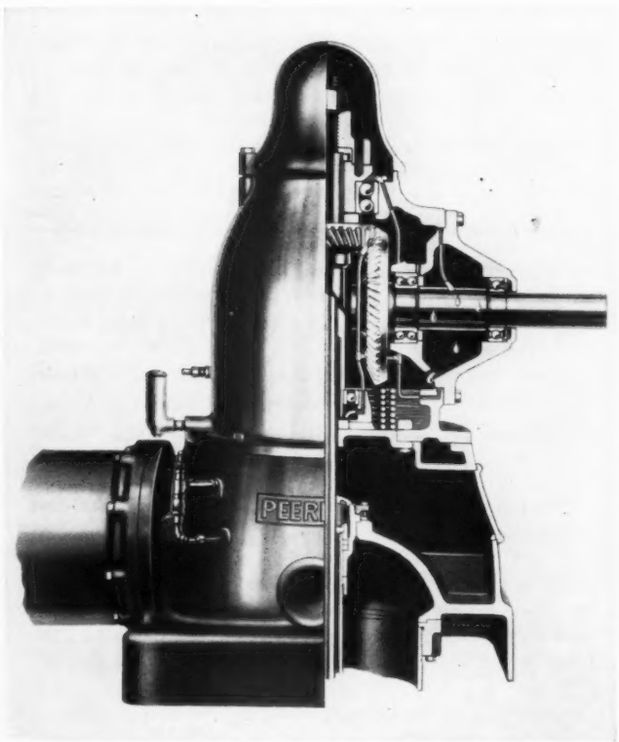


Drill head and table can be turned upside down for sanding large work. With a 1,740 r.p.m. motor the oscillator gives 200 working strokes per min., adjustable between ¼ in. and ⅞ in. Loosening the rocker arm thumbscrew and removing belt driving the Reciproc disengages it completely.

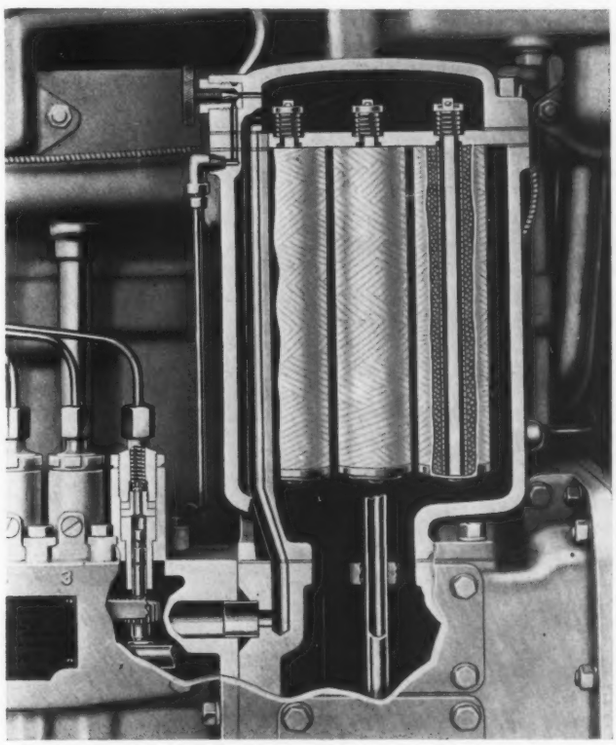
Improved Stuffing Box



Lantern rings in stuffing boxes are usually hard to pull out. For this reason, the engineers of the Mixing Equipment Company designed the stuffing box shown above, and used on their "lightnin" mixers such as the model shown to the right. After the upper packing has been removed the puller or extracting bolts are screwed into threaded holes in the lantern ring, which can then be pulled out without difficulty and without marring shaft.



Double-row ball bearings support the horizontal and vertical shafts of the Gearturbo head of Peerless Pump Company's deep-well turbine pumps. Extra heavy ball thrust bearings support lower end of geared shaft sleeve and outer end of horizontal shaft. Circulating oil lubricates unit.

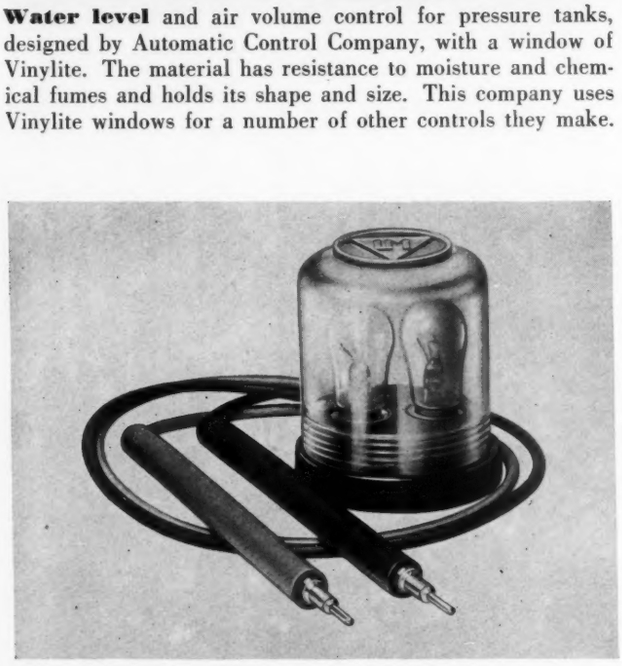


New filter designed by Caterpillar engineers for their diesel engines has cotton yarn wound on a metallic cylindrical screen wrapped with filter paper. Yarn is accurately spaced, with closer spacing toward core. Filter is easily removed; gage indicates when replacement of element is necessary.

MODERN DESIGNS – Transparent Plastics Applications

Water level and air volume control for pressure tanks, designed by Automatic Control Company, with a window of Vinylite. The material has resistance to moisture and chemical fumes and holds its shape and size. This company uses Vinylite windows for a number of other controls they make.

Another application for transparent plastics is found in the Arrow Beam flashlight made by Fulton Manufacturing Company. The barrels and caps are molded of Tenite in a variety of colors; the lens pieces are transparent Tenite. Injection molded by Fulton Manufacturing Company.



New test lamp for 110 and 220 volt circuits designed by Line Material Company has a transparent molded plastic hood screwed into molded plastic base. The two small lamps under the hood burn dimly at 110 volts, brightly at 220 volts. Flexible lead wires connecting to pointed prongs have 5-in. fiber handles. Transparent plastic "unbreakable" hood protects the lamps and also the operator should a lamp burst.

RANDOM JOTTINGS ABOUT NEW DEVELOPMENTS

SUPERFINISH increases the life of valve seats and disks twelvefold, according to Manning, Maxwell & Moore, engineers. The smooth high polish reduces friction and hence reduces wear.

STORAGE BATTERY for flashlights is announced by Ideal Commutator Dresser Company. Small in size, it fits all popular 2-cell, 1¼-in. size D flashlight cases. Electrolyte cannot spill, plates are more than ¼ in. thick; the battery withstands rough handling. A small charger—transformer and rectifier plates—recharges the battery in 12 hr. from a 110 volt, 60 cycle supply.

FORMING DIES in aircraft building are being made of cast zinc, with little machining required. Rubber made dies for embossing operations are also becoming popular. A block of rubber, incompressible, forces the sheet into the cavities of the steel female die.

BALL and ROLLER BEARINGS

Basic Factors Underlying Their Selection

G. A. UNGAR, *Consulting Engineer*
(Formerly Chief Engineer of SKF Ball Bearing Company, and author of the Bearings Section in Marks' *Mechanical Engineering Handbook*)

WHEN SELECTING a ball or roller bearing, the designer is likely to become confused by the wide differences in load capacity ratings for the same type and size bearing of different makes. It might be assumed that differences in accuracy of manufacture and materials used would account for the differences in load capacity ratings. However, as will be explained, the differences in ratings are caused primarily by the differences in the methods of establishing them, as adopted by manufacturers. An intelligent comparison of the load capacities of bearings of different makes therefore requires that the catalogued ratings be corrected to a standard of comparison that will be the same for all of them.

The load carrying capacity of rolling contact bearings, that is, ball, roller or needle bearings, if made of suitable material, accurately manufactured, correctly installed and housed and properly lubricated, is determined by the fatigue of the materials of races and rolling elements. The moment at which fatigue failure occurs is dependent upon the magnitude of the stresses imposed by the load and upon the number of stress applications. Manufacturers give the load rating of their bearings as pounds load capacity at a specified speed and specified hours of estimated bearing life.

Tests conducted under carefully controlled conditions have established the relation between load, speed and bearing life expectancy. When testing a number of bearings under definitely controlled conditions, the time when failure occurs is not the same for all bearings. Therefore, it is customary to use either the minimum life of one of a definite percentage of a large group of tested bearings or the average life of a certain percentage of a large group of tested bearings.

There is at present no agreement among manufacturers as to the ratio of expected life to tested life. Even the specified conditions of life testing are not the same for all manufacturers. The minimum life expectancy figures upon which the load ratings are based, are arrived at by SKF, Norma-Hoffmann, Timken, Tyson and Bantam by taking 90 per cent of the average test life of a large number of bearings tested under identical load and speed conditions and with lubricant renewed at regular intervals. Hyatt bearing ratings are based on an average life expectancy of 50 per cent, instead of 90 per cent, of the average test life of a large number of bearings tested under identical conditions but with only one initial lubrication charge and without lubricant removal.

When it is considered that the average life expectancy of a large group of bearings is from 3½ to 5 times their minimum life expectancy, it is obvious that the load ratings of the same size bearings based on average and minimum life expectancy will be widely different and therefore cannot be used as a basis of direct comparison.

Another fact to be considered when comparing published rated load capacities is that there is no uniform standard as to the total number of hours of bearing life, be it average or minimum, upon which the rated capacity is based. Bantam uses 3,000 hr., SKF 2,500 hr., Fafnir and New Departure 3,500 hr., Marlin Rockwell and Hyatt 5,000 hr., Norma-Hoffmann 10,000 hr. average life. Timken and Tyson base their ratings on 3,000 hr. minimum life, instead of basing them on an average life figure.

To illustrate the effect of the differences in life expectancy figures used for the basis of rating bearings, Fig. 1 charts the catalog ratings at different speeds for ball bearings of three different manufacturers. Each bearing has the same number of balls of the same diameter and is made of the same material. SAE 52100, heat-treated to an average Rockwell C hardness of 62 is used for races and balls and all parts are precision finished. The actual tested or service life will be the same for each of the bearings because they are identical in all respects but, as shown in Fig. 1, their load ratings or rated capacities are considerably different. The difference is caused entirely by the difference in the arbitrarily

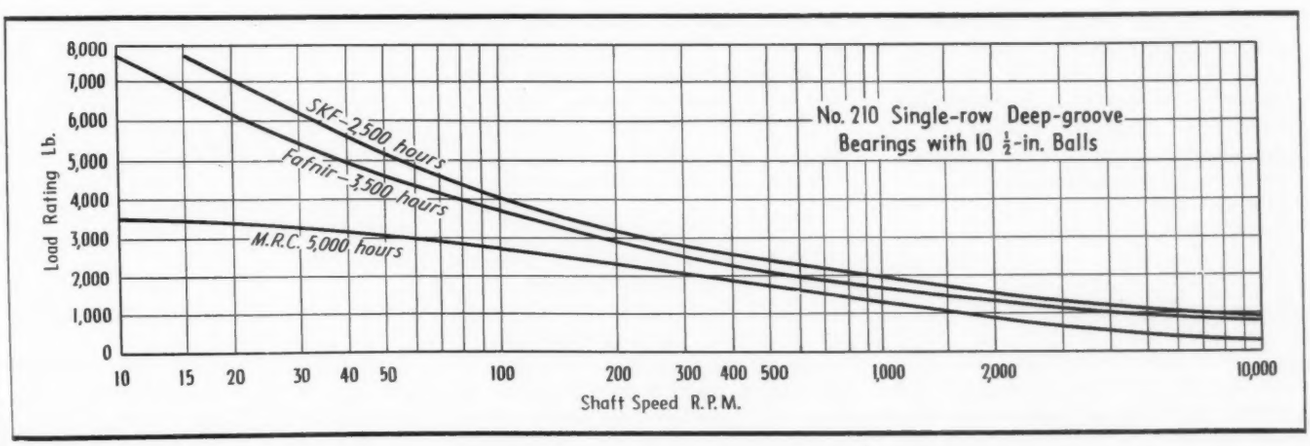


Fig. 1—Graph showing how difference in arbitrarily chosen average bearing life of three manufacturers affects catalog ratings

chosen number of hours of average bearing life.

From the foregoing it is evident that the published material available to the designer for the selection of rolling contact bearings must be used with considerable discretion in spite of the fact that manufacturers have made available a great mass of accurate and valuable data. The following considerations and procedures, based on information taken from catalogs and engineering publications of the different manufacturers, will enable the designer to make a fair comparison of the actual load capacities of different makes of bearings and the criteria or rules by which to arrive at a correct judgment in the selection of bearings.

Actual Bearing Life

From data obtained from innumerable tests conducted by different bearing manufacturers, the following empirical equation was developed for calculating the actual number of hours of life of a radial or angular bearing:

$$H = \frac{C}{NP^x}$$

where

H = hours of bearing life, actual
N = r.p.m.
P = load in lb., actual radial load in the case of a radial bearing. For radial and thrust loads, P is calculated as the radial load equivalent to the combined radial and thrust load
C = a constant which varies with the type and size of the bearings
x = exponent determined by the manufacturer by tests. For SKF bearings $x = 3$, for Marlin Rockwell $x = 3.3$, for Tyson roller bearings $x = 3.33$

The actual hours of expected bearing life H can easily be calculated from H_c, the average life of a bearing as rated by the manufacturer, and P_c, the catalog load rating in lb., for any average total bearing load P_{av} from the equation

$$H = H_c F^x \quad \text{where} \quad F = \frac{P_c}{P_{av}}$$

The "safety factor" F is based on data obtained from the service experience of the manufacturer. The exponent x is the same as in the formula for H given above. For convenience in applying the above formula, the curves for five bearing manufacturers in Fig. 2 give the average hours expected life directly from the multiplying factor $(P_c/P_{av})^x$.

Load Capacity

In order to evaluate the load carrying capacity of a bearing for a given application, it is advisable to consider both the load carrying capacity at the given r.p.m. and the space occupied by the bearing. The load criterion is therefore expressed in lb. per sq. in. of projected bearing area, the projected bearing area being the area projected by the outer race upon a plane passing through the axis of the bearing, minus the similarly projected area of the bore or the shaft diameter if the rolling elements operate directly on the surface of the shaft.

The values of load capacity per unit of projected area are tabulated in the Reference Book Sheet on page 47. The load ratings as given by the manufacturers were converted to the equivalent rating at 300 r.p.m. and a minimum life expectancy of 3,000 hr. These calculated load capacities were then divided by the projected bearing area as defined above, thus obtaining the load capacity factor as tabulated. These load factors are averages for bearings ranging from ½ in. to 4 in. bore or shaft diameter, unless otherwise stated in the table.

The table gives either the radial or the axial load factor, and a factor X which applies to the formula:

$$\textit{Radial Load} = X \times \textit{Axial Load}$$

The ability of a bearing to withstand shock is rated as the product of the calculated bearing load and the shock safety factor recommended by the manufacturer. The table on page 47 gives these shock factors for three degrees of shock: (1) steady load, (2) moderate shock, and (3) extreme and indeterminate shock. Bearings with greater shock resisting ability will have a numerically higher shock factor.

Obviously, three factors are involved in the catalogued bearing capacity ratings, namely, the load, the speed in r.p.m., and the hours of operation. In considering load capacities, it is necessary to distinguish between static and dynamic loads. The permissible static load capacity of a bearing, known as the non-brinelling static load, is that load which is reached just before a local permanent surface deformation takes place. Dynamic load capacities of a rolling contact bearing are substantially lower than the static load capacity because of the effect of fatigue.

Ratings of radial and angular bearings are always based on revolving inner races. Here the number of stress reversals is less than for outer races revolving at the same speed. The life expectancy of bearings with revolving outer races is therefore less than that of the same bearing with revolving inner race, amount of difference depending upon type and size. Conversion factors for determining ratings for revolving outer races are supplied by the manufacturers.

Maximum Recommended Speed

Of importance in high-speed applications approaching the maximum speeds recommended by the manufacturer, the table on page 47 tabulates maximum allowable speed in hundreds of r.p.m.

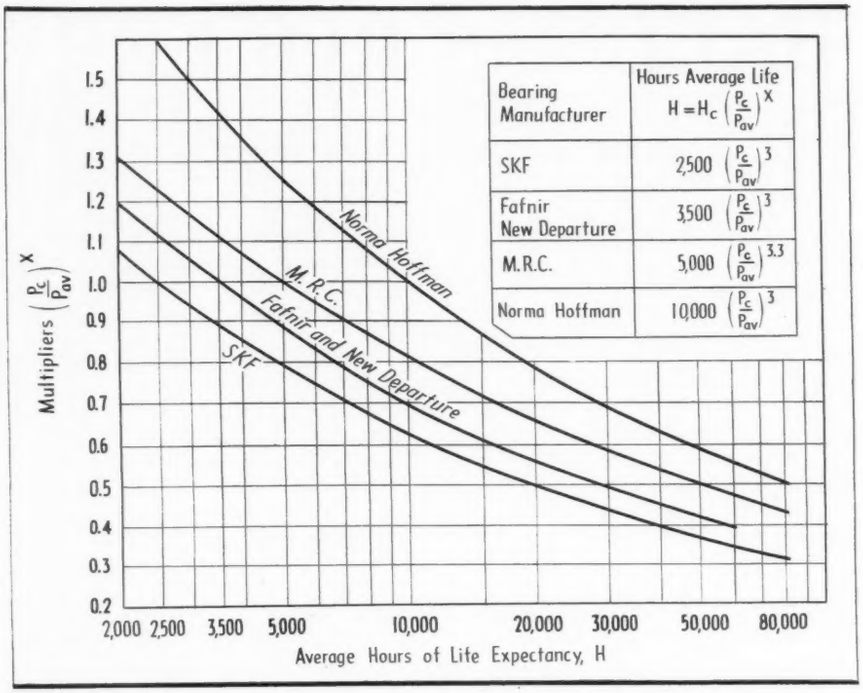


Fig. 2—Chart for determining average hours expected life of a bearing from catalog life and load ratings, when the average service load differs from the catalog load

for bearings of three grades of finish: regular bearing finish (reg.), precision finish (prec.), and ultra-precision finish (u.prec.). These speeds for bearings with rotating inner races are tabulated for ½ in., 1 in., 2 in., and 4 in. bore shaft diameter. For bearings with standard SAE dimensions the bore of the light series bearings is used while for the medium and heavy series bearings the bore of the light series bearing corresponding to the same outside diameter were used.

Bearing friction is of importance particularly in higher speed applications. Heat generated is a function of frictional resistance and speed. High friction might create problems of heat transfer in order to avoid excessive temperatures. In view of the many factors affecting actual frictional values the friction criteria of the table (p. 47) express only relative magnitudes which are expressed as low (1), medium low (2), medium (3), medium high (4), and high (5).

Relative values of bearing noise are expressed by the acoustic criterion tabulated in the table (p. 47), since actual decibel readings will depend on a number of factors not actually functions of the bearing proper. The values chosen for comparison are: silent (1), fairly silent (2), and noisy (3), at bearing speeds varying from approximately one-third to one-half the maximum r.p.m. up to the maximum r.p.m. of the bearing type and size in question.

The ability of the bearing to compensate within itself for axial misalignment and shaft deflection is expressed in the table (p. 47) as permitting: extreme misalignment up to plus or minus 5 deg., slight misalignment up to plus or minus 1 deg. and no misalignment permissible. Spherical types, or self-aligning bearings, permit extreme misalignment. The number of bearings (1 or 2) of the type in question required to provide axial stabilization of the shaft on which the bearings are mounted is also listed in the table.

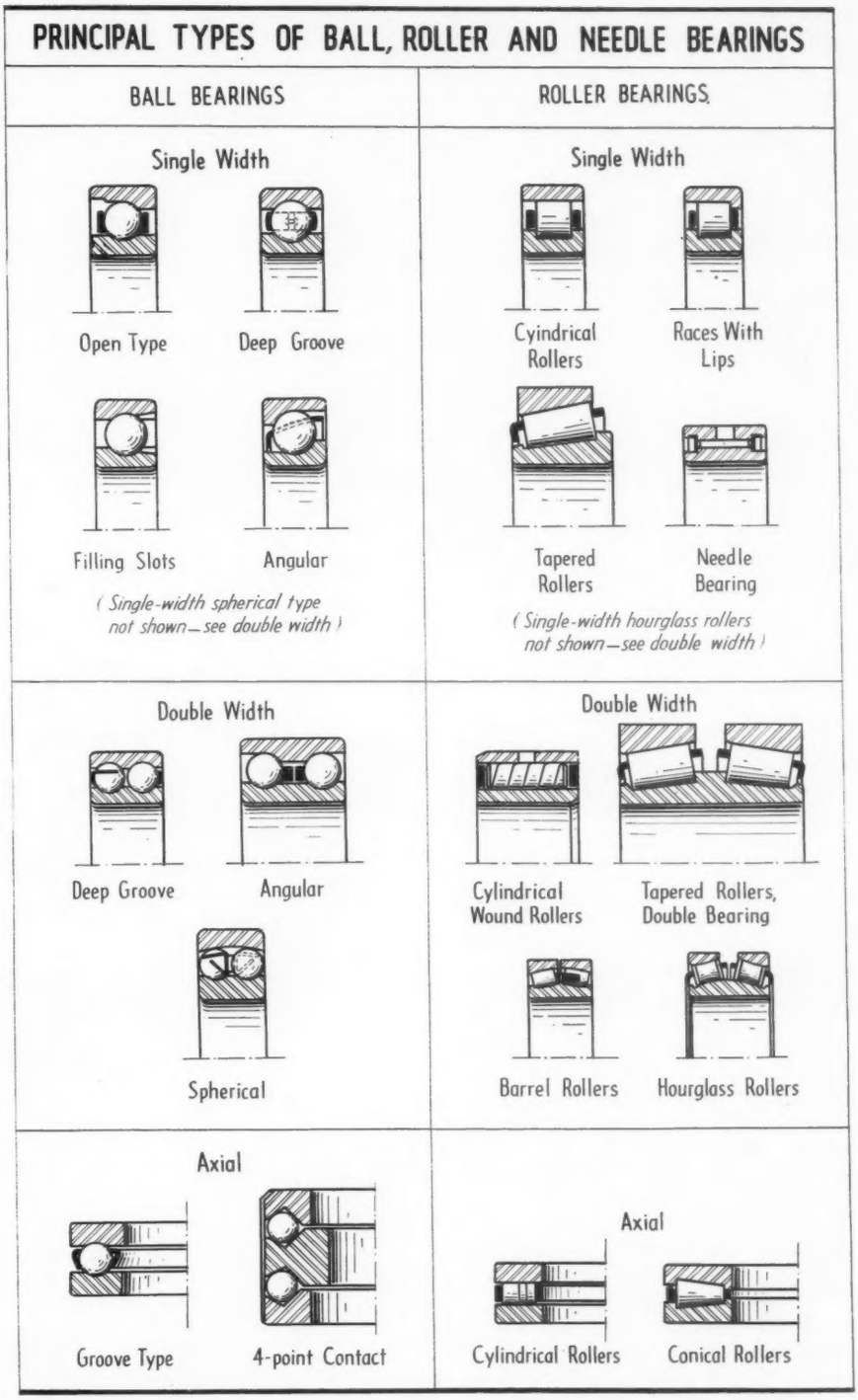
Considerations Not Tabulated

The foregoing criteria lend themselves to general tabulation but there are additional factors which must be considered when making a selection of bearings which is to provide a really economical installation. The most important of these is cost.

When selecting bearings on the basis of cost, the correct cost comparison is, of course, the total cost divided by the number of hours of life expectancy. The total bearing cost is the sum of the first cost, the cost of the bearing mounting (machining cost of shaft and housing) and of the cost of maintenance (lubrication) during the estimated bearing life.

Another important consideration is that of ready replacement. Designers should specify bearings of a type and make that is certain to be in dealers' stocks wherever the ultimate geographical location of the machine may be.

The wear of associated parts should also be properly considered. The associated parts primarily affected by the wear of rolling contact bearings are shafts and housings. Bearings which are complete units consisting of inner and outer races with rolling elements have little effect upon shaft and housing surfaces when bearing wear takes place, but where shafts or housings fulfill the functions of raceways, then wear becomes a more important factor in the cost of possible replacements and repairs. For this reason shafts and housings are used as raceways primarily only for reasons of radial space economy, and are not so used if available.



HYDRAULIC COUPLINGS–I

Operating Principles, Performance Characteristics

R. D. McCAUSLAND
Hydraulic Coupling Division, American Blower Corporation

WIDESPREAD attention is being given to fluid couplings today not only because of their ever-increasing use in automobiles, pioneered in the United States by Chrysler two years ago, but because these couplings are being applied to bus, truck, road machinery, railroad and other industrial machinery for both constant-speed and variable-speed transmission. This article will explain the basic principles of operation of traction-type, constant-speed, and scoop-tube, variable-speed-type hydraulic couplings, and will set forth the inherent advantages in fluid drives and the scope and limitations of their applications. Several different types have been developed to meet specific requirements of different classes of machinery drives.

The hydraulic coupling operates on a hydrokinetic or turbo-principle in which power is transmitted from the driving rotor to driven rotor by kinetic energy continuously supplied to a circulating liquid. The simplest form of hydraulic coupling, the traction type, consists essentially of three main elements: the primary element or impeller, the secondary element or runner, and the inclosing cover or casing.

In Fig. 1 is shown an exceptionally large runner, 6 ft. 10 in. outside diameter, fabricated by Lukenweld, Inc. entirely out of welded steel plate. The radial vanes or ribs guide the circulation of the hydraulic medium. The vanes in the half-circle section midway between hub and rib are a detail of mechanical design. They are usually omitted in smaller couplings as can be seen in Fig. 2, which is a disassembled view of a traction coupling. The impeller and runner differ primarily in the number of radial vanes extending from the hub to the rim. When assembled, the impeller is mounted on the driving shaft or flywheel, the runner is keyed to the driven shaft. There is a small clearance between runner and impeller. An oil sealed casing envelopes the two units. The impeller usually serves as one-half of the split casing.

The runner, since it has no torsionally rigid connection to the impeller of the coupling, figuratively "floats" in the working medium which is usually a light mineral oil having non-corrosive and lubricating properties. Assuming the coupling to be full of oil, when the impeller is turned over, centrifugal force acts on the liquid between the vertical vanes, tending to throw the oil outward from the inner diameter as indicated by arrows, Fig. 4. Similarly, centrifugal force acting on the oil in the runner, tends to throw it radially outward. The speed of the impeller is greater than that of the runner, and hence the radial pressures exerted by centrifugal force are greater in the impeller than in the runner. This causes oil circulation, as indicated.

The power transmitting capacity of a hydraulic coupling is dependent upon

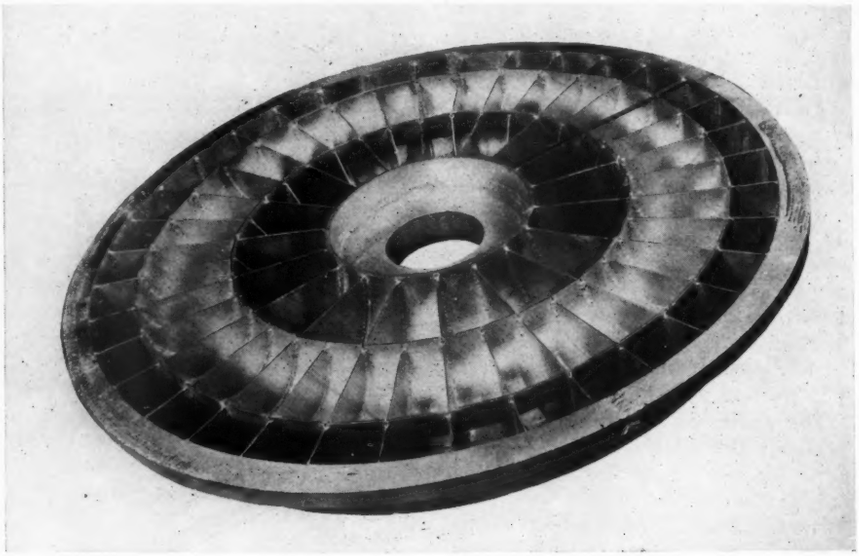


Fig. 1—Almost 5 ft. in dia., this runner weighs 2,920 lb. and is constructed entirely of roller steel pieces welded together; built by Lukenweld, Inc. The impeller that goes with it is almost identical except for the number of radial vanes for guiding the oil

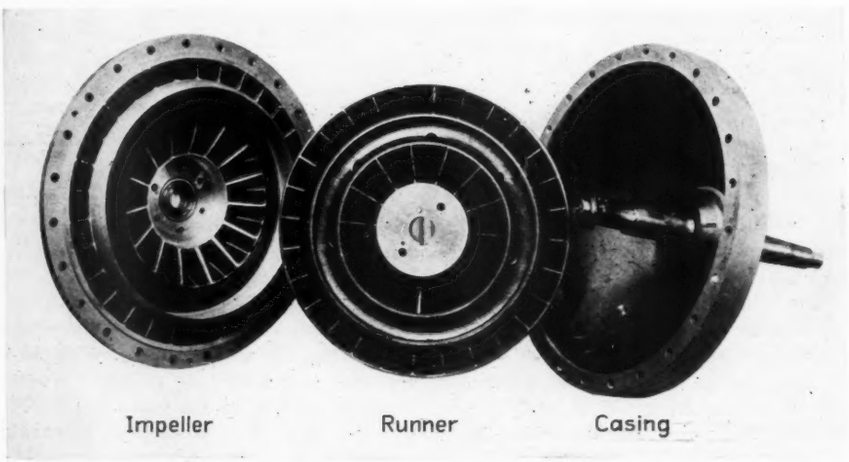


Fig. 2—Disassembled traction coupling. The impeller provides one-half of the casing. The casing shown rotates with the impeller, being bolted to it

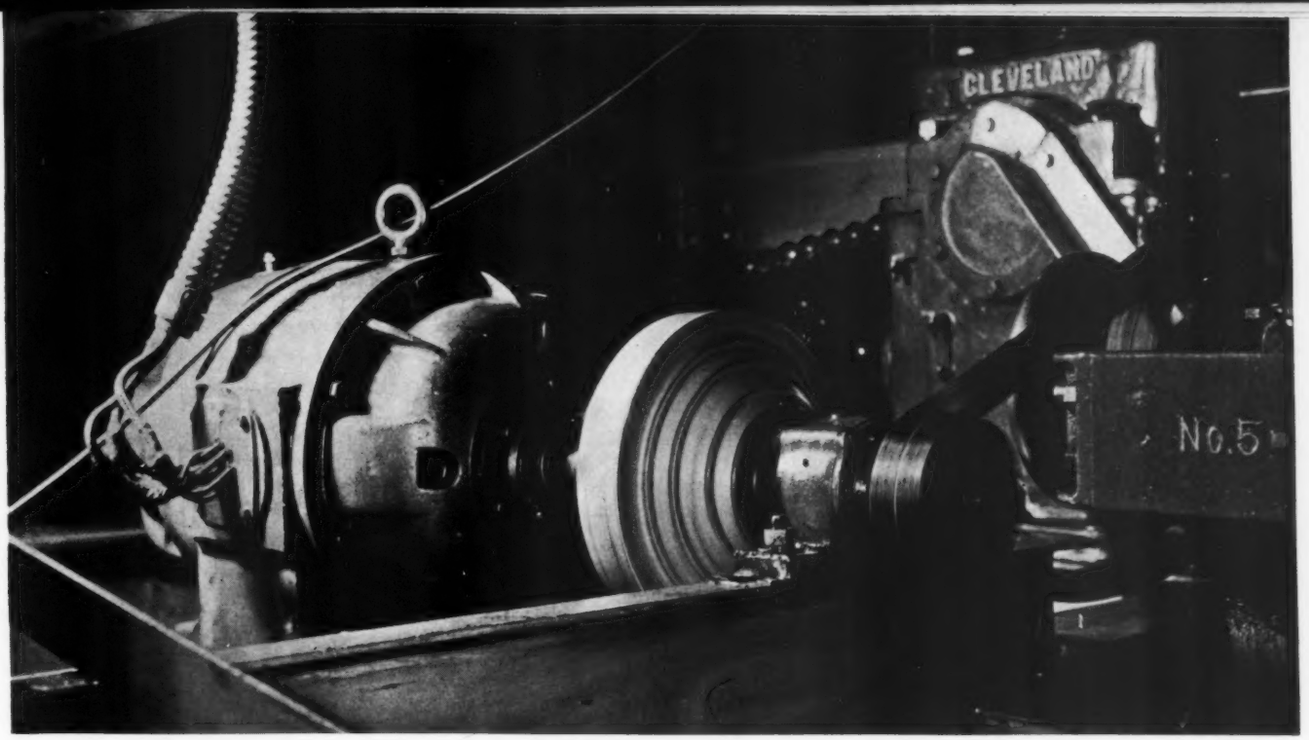


Fig. 3—Hydraulic couplings for conveyor drives protect the equipment from possible damage when the conveyor jams

the conversion of mechanical power delivered to the impeller, into kinetic energy which is imparted to the circulating oil as it passes through the impeller. When the oil subsequently passes through the runner its kinetic energy is reconverted into mechanical energy. Although the following explanation is not complete and accurate, it serves to give a conception of the principle of operation of fluid couplings. The kinetic energy of the circulating oil may be expressed in terms of $\frac{1}{2}MV^2$ where M is the mass of the liquid in motion and velocity V is the resultant of the velocity of circulation and the tangential velocity of the rotating mass of oil. Thus, the kinetic energy of every particle of the circulating oil is a function of its absolute velocity which the resultant of the radial velocity and the velocity tangential to the circular path of rotation. The radial velocity changes must be inversely proportional to the cross-sectional areas of the path of flow and hence are relatively small and can be ignored. The tangential velocity component will be directly proportional to the r.p.m. speed and the distance from the center of rotation. Only the kinetic energy associated with the tangential velocity need be considered.

When a slug of oil passes from the impeller to the runner, it will have a certain tangential velocity V_t as determined by the linear velocity of the impeller at the point where the slug of oil leaves it to enter the runner. The circulating oil will then travel radially toward the center of the runner. When the oil first entered the impeller it had a kinetic energy of the order of $\frac{1}{2}MV_t^2$ in the tangential direction. As the oil moves radially toward the center its tangential velocity decreases proportional to the distance from the center of rotation, decreasing to a velocity of V_o at a point near the hub where the oil leaves the runner to return to the impeller with a kinetic energy of the order of $\frac{1}{2}MV_o^2$. The difference between these kinetic energies $\frac{1}{2}M(V_t^2 - V_o^2)$ has been absorbed by the runner and appears as power output of the coupling.

A simple mechanical analogy of the operating principle of hydraulic couplings is illustrated by a person sitting on a piano stool with arms outstretched and holding a heavy weight in each hand. If the person is now spun slowly and then suddenly pulls the weights close to his side, his speed of spinning will greatly increase.

It is obvious that there must be slip between the impeller and the runner in order to provide the unbalanced centrifugal forces that cause circulation. If both runner and impeller rotated at the same r.p.m., the centrifugal force acting on the oil would be the same for both runner and impeller, and hence there would be no unbalanced

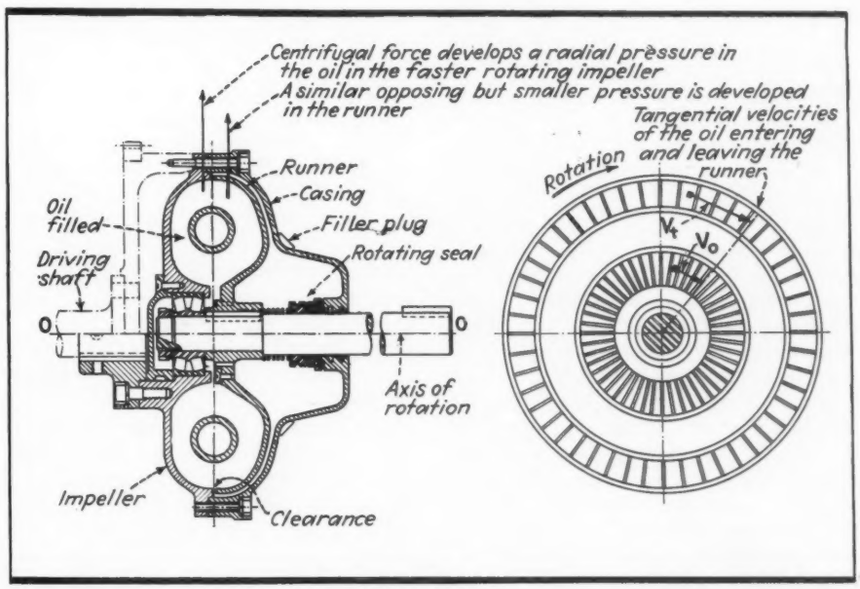


Fig. 4—Schematic diagram showing how the hydraulic oil is circulated by centrifugal force and gives up a portion of its kinetic energy to the impeller as the oil travels from the rim towards the hub

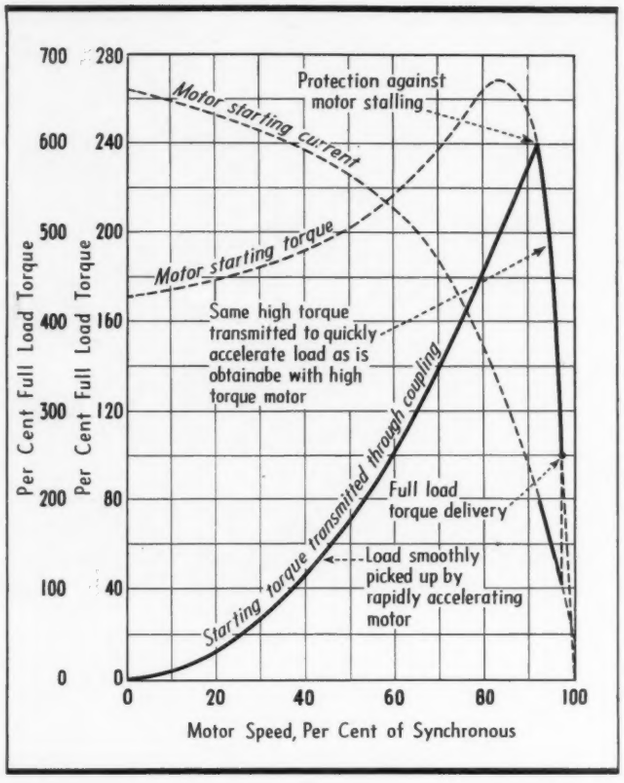

Protection against motor stalling
Motor starting current
Motor starting torque
Same high torque transmitted to quickly accelerate load as is obtainabe with high torque motor
Starting torque transmitted through coupling
Full load torque delivery
Load smoothly picked up by rapidly accelerating motor
Per Cent Full Load Torque
Per Cent Full Load Torque
Motor Speed, Per Cent of Synchronous

Fig. 5—Performance curve of traction-type hydraulic coupling with general-purpose, squirrel-cage motor as driving unit

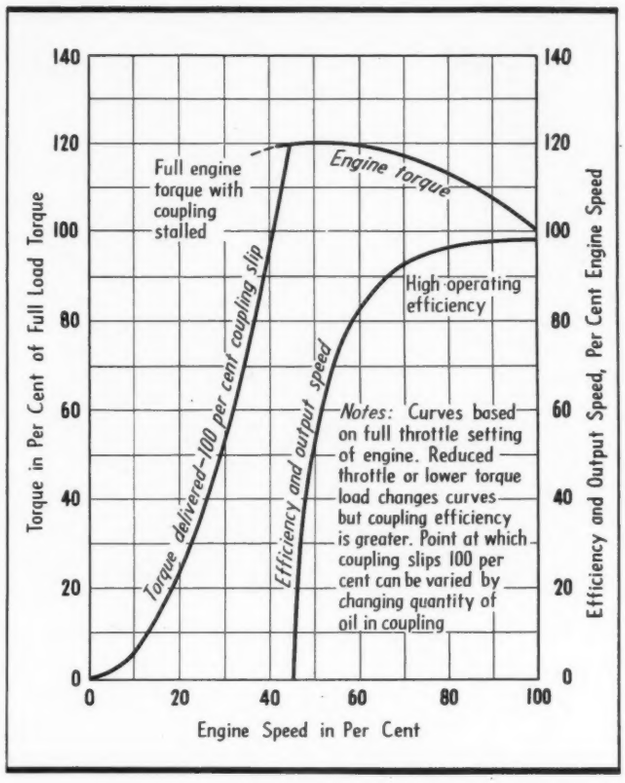

Full engine torque with coupling stalled
Engine torque
High operating efficiency
Torque delivered-100 per cent coupling slip
Efficiency and output speed
Notes: Curves based on full throttle setting of engine. Reduced throttle or lower torque load changes curves but coupling efficiency is greater. Point at which coupling slips 100 per cent can be varied by changing quantity of oil in coupling
Torque in Per Cent of Full Load Torque
Efficiency and Output Speed, Per Cent Engine Speed
Engine Speed in Per Cent

Fig. 6—Performance curve of traction-type hydraulic coupling with an internal combustion engine as driving unit

pressure to cause circulation. Without circulation of the oil, no torque can be transmitted. Generally, the slip is only 2 per cent, or even less, although in some special cases the slip may be of the order of 3 or 4 per cent, with satisfactory operation. The limit of torque transmission is reached when the slip is so great that the oil circulation becomes turbulent and the oil churns. This behavior makes impossible the stalling of the motor or engine which is driving the coupling.

It is obvious that the horsepower transmitted by a fluid coupling will be a function of the diameters of the coupling, the density of the hydraulic fluid, the rate of its circulation, the r.p.m. of the coupling, and the per cent of slip. The power transmitted by a fluid coupling is proportional to the density of the liquid used as a medium and considerable effort has been expended by numerous oil companies in search of a special-type non-corrosive liquid having lubricating qualities as well as low viscosity, low freezing point, high boiling point, freedom from chemical breakdown when subjected to the combined effects of time, temperature and turbulence, and availability at a reasonable price.

Mercury, for instance, is about fifteen times as heavy as oil but it is destructive to non-ferrous materials in the coupling. Its weight and cost more than offset the gain in reduced coupling size. Best results have been obtained by the use of a good grade light mineral oil such as is commonly used in steam turbines and having a viscosity of about 150 Saybolt second units at 100 deg. F. This oil has a low freezing point; thus all types of fluid couplings can be used out-of-doors where extreme weather conditions prevail.

Temperature of the oil does not have a great deal of effect on coupling performance. However, a rise in oil temperature caused, for instance, by the coupling being operated under load at a high slip will improve the efficiency of the coupling resulting in slightly higher torque being delivered. This may be contrary to first impressions in that a rise in temperature will lower the viscosity and density, but this is more than offset by the increased rate of circulation due to reduced resistance to flow.

The quality of construction of a hydraulic coupling is an important factor in determining efficiency. The rotating members of the coupling may either be made from sand castings, die castings, fabricated welded steel or machined from a solid block. The factors influencing the kind of construction are: (1) allowable weights and inertia, (2) cost per unit of production rate and (3) efficiency, influenced by the roughness of the surface in contact with the circulating oil and the thickness of the vanes in the runner and the impeller. Where mass production is not warranted, sand castings carefully ground to insure a smooth surface are commonly used.

The manner of mounting hydraulic couplings in various drives is dependent upon actual conditions found with each application. However, two designs, hub and flywheel mounting, have been adopted. In the former design the impeller of the coupling is constructed with a cored out hub which will slip onto a projecting shaft. This method is used for mounting the coupling directly on a motor shaft or on a supported jackshaft. The impeller of the coupling can also be designed so that it can be bolted directly to the flange of the flywheel. This method is commonly used with internal combustion engines having the coupling mounted directly on the engine flywheel.

The vortex action of the oil circulating between the coupling impeller and runner creates a thrust between these two elements varying in direction and magnitude and depending on load and slip conditions. This thrust is absorbed internally in the traction-type and scoop-control-type hydraulic couplings. In the pump-controlled type, it is usu-

ally taken by an outboard bearing.

There have been many oil seals designed for sealing between shaft and casing, mostly of the diaphragm type, each having various degrees of success. Relatively recent research work has resulted in the development of a special contact seal assembly which has proved satisfactory. The variable-speed type hydraulic couplings do not present any oil seal problems that cannot easily be taken care of by the use of simple labyrinth glands.

The power loss in fluid couplings is practically entirely a function of frictional resistance to circulatory flow of the liquid. This resistance is dependent upon the design of the coupling circuit or curvature of the walls, the smoothness of the walls and vanes, the amount of space taken up by the vanes, internal flowing friction, and viscosity and density of the liquid used which are in turn affected by the temperature and degree of aeration.

If the load on a hydraulic coupling running at constant speed is increased, slip increases, resulting in a faster circulatory liquid flow. As stated above, with increasing slip a point will be reached where turbulent flow begins. This will decrease the rate of oil circulation which in turn will further decrease the torque transmitted, increase the slip, and finally the coupling will stall. In many applications this is desired. By varying the quantity of liquid within the coupling and by introducing various degrees of baffle to impede the high velocity circulating liquid, the overload torque at which the coupling will stall can be regulated. One method is to design the coupling with a baffle or disk in the path of flow of the circulating oil, positioning the baffle in such a way as to disrupt the flow when the velocity of circulation has reached the point corresponding to the maximum torque it is desired to transmit. The design of the baffle is such that it does not interfere with circulation under normal slip full-load and full-speed operation. The baffle also reduces the drag torque through the coupling at low driving or idling speeds. In some cases the use of a large baffle to reduce drag torque has eliminated the need for a disconnecting clutch in the installation.

From the preceding description it can be seen that the fluid coupling allows any prime mover to be started up under no load condition and then gradually and smoothly pick up the load as it gains speed. This is shown in the performance curves of Fig. 5 for an electric motor drive with fluid coupling transmission. Referring to Fig. 5, the starting or stalled torque curve of the coupling shows that any motor driving a load through a hydraulic coupling starts under no-load and as the motor accelerates under its own torque it applies a gradually increasing torque to the load, making a shockless start and smooth acceleration. Little time is lost in this start because the motor attains almost full-load speed in a split second and then it transmits its peak torque, or less if desirable, to quickly accelerate the load.

The starting torque delivered to the load by a standard, general purpose, squirrel cage motor driving through a fluid coupling is comparable to that of a high starting torque, high slip motor which is more expensive and less efficient. Furthermore, the starting current drawn by the normal torque motor with fluid coupling is only a fraction of what it would be with the motor direct connected. When the driven machine is up to full speed and full load the coupling has normally only one to a maximum of four per cent slip, which approximates the loss of power in the coupling. However the slip increases if an overload occurs and will increase to 100 per cent if the overload reaches a predetermined value. Thus the coupling protects the motor from any damaging overload and also limits the amount of torque transmitted to the stalled load. The motor can be reversed while the driven machine is still running and thus apply a smooth and gradually increasing braking effect.

The performance of the fluid coupling used with an internal combustion engine is shown in Fig. 6. The engine starts against no-load and a smooth take-up of the load is obtained by simply accelerating the engine. The initial high slip in the coupling rapidly decreases and falls to around 1 per cent to 3 per cent in the normal range of speed and load. Referring to the curve, it can be seen that it is impossible to stall the engine however much the load is increased, since the effect of overloading the coupling merely pulls down the speed and a point is reached when the slip in the coupling rises to 100 per cent, that is, the coupling stalls. Under these conditions the output shaft is stalled while the engine continues to run at a reduced speed, yet full engine torque is delivered to the load. There is no loss or gain of torque in the coupling when stalled or under any operating conditions, and when the overload is removed the coupling will automatically speed up—like a stalled steam turbine.

The cushioning effect of fluid drive absorbs shock loads and torsional vibration that would tend to be transmitted from the engine to the driven machinery and vice versa. The softness of fluid drive and the slippage characteristic of the coupling reduces wear on friction clutches and takes the shock from gear and chain drives, extending their life. Flywheel energy can be transmitted through the coupling without any shock to the equipment simply by running the engine at top speed and dropping in a friction clutch positioned between engine and fluid coupling. Another inherent characteristic of fluid couplings is that due to their increased slippage at overloads they automatically equalize the load between two or more engines or motors operating in tandem at approximately the same speed. This feature eliminates the common difficulty found in attempting to synchronize prime movers driving in tandem.

Editor's Note: The design and application of hydraulic clutches for variable-speed transmission will be covered in February PRODUCT ENGINEERING.

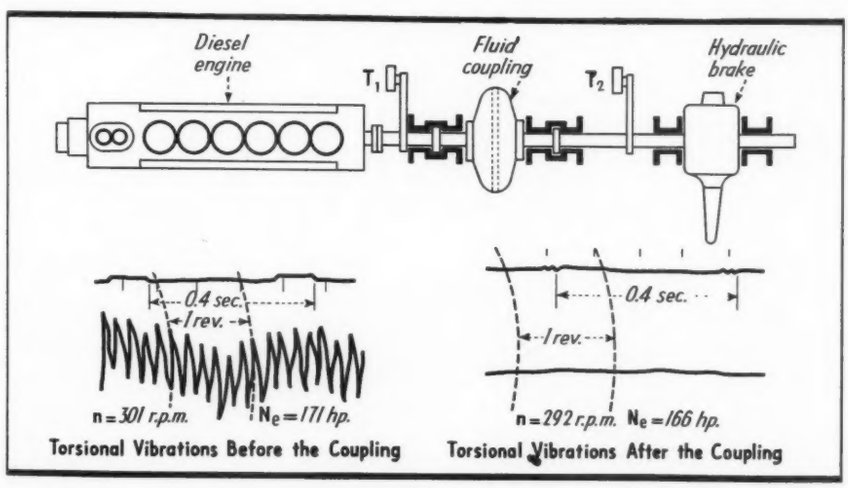


Fig. 7—Test set-up and charts made by vibrographs showing how the fluid coupling dampens torsional vibrations

CRITICAL SPEEDS OF SHAFTS

MERHYLE F. SPOTTS
Johns Hopkins University

THE numerical computation of critical speeds of shafts of variable diameter is generally laborious. The technique explained here will often effect an appreciable saving in the labor required for such work.

Critical speed of a shaft is that r.p.m. at which the rotating system becomes unstable and vibrations of large amplitude build up. This condition of resonance occurs whenever the shaft is rotating at a speed equal to its frequency for free lateral vibration. It is demonstrated in treatises on vibration ("Vibration Problems in Engineering," Timoshenko, Second Edition, 1937, p.94, D. Van Nostrand Co.) that the frequency for the lowest mode of motion is given by the equation,

$$f = \frac{1}{2\pi}\sqrt{\frac{g\Sigma W_i y_i}{\Sigma W_i y_i^2}} \text{ cycles/sec.} \qquad (1)$$

where W_i are the loads on the shaft, y_i are their respective static deflections, and g is the gravitational constant. The summation is to extend over all the weights on the shaft.

Thus it is necessary to have the static deflection at each load, and when the shaft is not of uniform diameter considerable labor may be involved. For such shafts it is sometimes convenient to make use of the equation of elastic energy (Elastic Energy Theory, J. A. Van den Broek, p.42, John Wiley & Sons, 1931) for the deflection of a beam,

$$y = \int \frac{M_p M_g dx}{EI} \qquad (2)$$

Where M_g is the bending moment resulting from the applied loading, M_p is the bending moment caused by a 1-lb. auxiliary load acting at the point and in the direction that the deflection is desired, E is the elastic modulus for the material and I the moment of inertia of the cross-section. The integration is to extend over entire length of shaft.

An efficient means of carrying out this integration is to give Equation (2) a geometric interpretation. Referring to Fig. 1, let the product $M_p\ M_g/I$ be considered one of the rectangular end surfaces of the solid represented therein. If $M'_p\ M'_g/I$ is the other end, and if these surfaces are parallel and separated by a perpendicular distance dx, the volume of this solid of length dx, when divided by E, will be the integrand of Equation (2). Such solids are considered as being located throughout the length of the shaft, the sum of which gives the value of the integral or, by Equation (2), deflection y can be calculated.

For the problems here considered, the bending moment diagrams will be represented by segments of straight lines. Therefore, if, instead of the infinitesimal dx, solids having finite lengths are considered, these volumes may then be computed by elementary geometry and the deflections found.

The prismoidal equation,

$$\text{Vol.} = \frac{l}{6}(A + 4M + B) \qquad (3)$$

is convenient for such purposes. Here A and B are the areas of the end surfaces and M is the area of the cross-section mid-way between.

The shaft must be divided into segments in such a manner that the shaft has a uniform diameter throughout each interval. Otherwise there will be a discontinuity in the curve representing M_g/I. When a shaft carries a uniform load, the bending moment may be closely approximated by assuming that the shaft is divided into small intervals with the load on each interval being considered as concentrated at its center. Such a shaft is shown in Fig. 2(a), which supports a load of 100 lb. per in. over the central section which is 27 in. long and 3 in. in dia. This central portion is divided into three equal lengths of 9 in. each with a load of 900 lb. at their respective mid-points.

The bending moment diagram Fig. 2(b) is everywhere divided by the corresponding moment of inertia of the shaft giving diagram (c) of Fig. 2. For the deflection at the center, place the 1-lb. auxiliary load at E, Fig. 2(d), which gives the M_p diagram Fig. 2(e). The result of multiplying M_g/I by M_p is also shown in Fig. 2(e), tabulated below the base of the diagram.

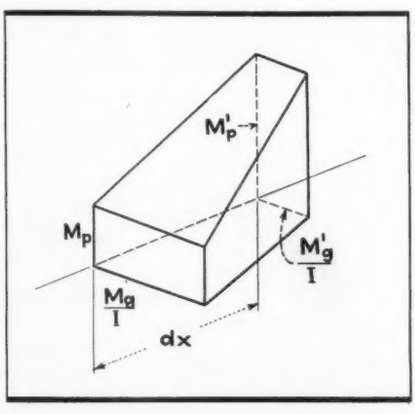


Fig. 1—Geometrical interpretation for carrying out integration of Equation (2)

The integrand volumes may now be computed. For the portion AB, the solid would be the pyramid with area of base 30,180, ($M_p\ M_g/I$) altitude 7½ in., dx and volume ⅓x7½x30,180 = 75,450. For BC there results the prismoidal solid of end areas 13,530, and 65,460, and mid-section 34,620, and length 9 in. The volume would be 1/6x9(13,530+4x34,620+65,460) = 326,205. Dividing these values by E, the modulus of elasticity gives the integrands. The complete calculations for the deflection y at M are given below. Due to symmetry, it is only necessary to make computations for one-half the shaft; the result is then doubled.

$\frac{dx}{\text{sec.}}$	$M_g M_p dx/I$	
AB	⅓ × 7½ × 30,180	= 75,450
BC	⅙ × 9(13,530 + 4 × 34,620 + 65,460)	= 326,205
CD	⅙ × 4½ (46,220 + 4 × 59,680 + 74,860)	= 269,850
DE	⅙ × 9 (74,860 + 4 × 97,400 + 122,240)	= 880,050

For one-half the shaft, $\Sigma M_g M_p / dx/I$ = 1,551,555

For mid-point of shaft,

$$y_e = \frac{2 \times 1{,}551{,}555}{30{,}000{,}000} = 0.1034 \text{ in.}$$

For the deflection at D, place the auxiliary load as shown in Fig. 2(f). The M_p diagram together with the values of $M_g\ M_p/I$ are shown in Fig. 2(g). Computations for y_d are given below.

$M_g M_p dx/I$		
⅓ × 7½ × 39,230	=	98,075
⅙ × 9 (17,580 + 4 × 45,010 + 85,090)	=	424,065
⅙ × 4½ (60,080 + 4 × 77,590 + 97,320)	=	350,820
⅙ × 9 (97,320 + 4 × 92,240 + 85,560)	=	827,760
⅙ × 9 (85,560 + 4 × 68,180 + 52,410)	=	616,035
⅙ × 4½ (52,410 + 4 × 41,780 + 32,350)	=	188,910
⅙ × 9 (45,820 + 4 × 24,230 + 9,470)	=	228,315
⅓ × 7½ × 21,130	=	52,825
$\Sigma M_g M_p dx/I$	=	2,786,805

$$y_d = \frac{2{,}786{,}805}{30{,}000{,}000} = 0.0929 \text{ in.}$$

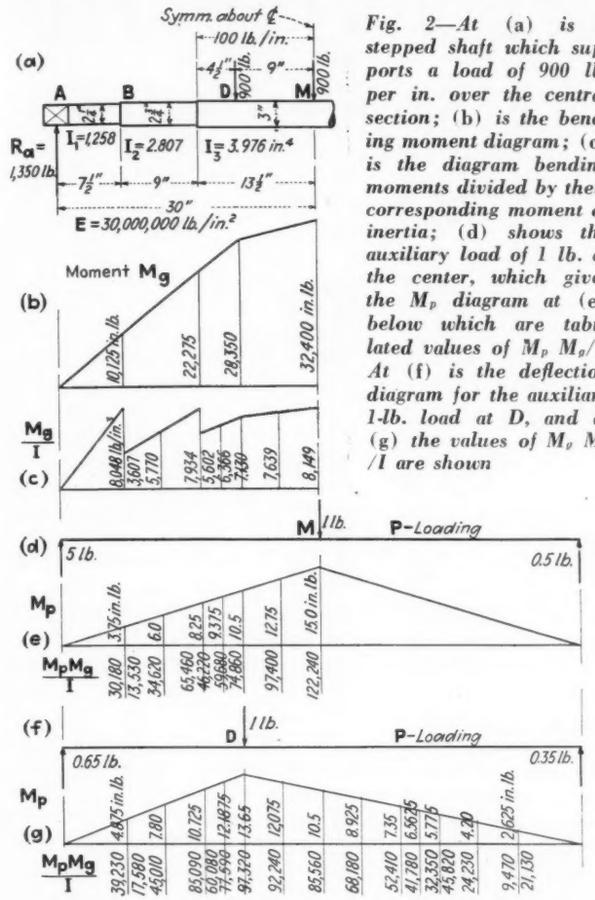


Fig. 2—At (a) is a stepped shaft which supports a load of 900 lb. per in. over the central section; (b) is the bending moment diagram; (c) is the diagram bending moments divided by their corresponding moment of inertia; (d) shows the auxiliary load of 1 lb. at the center, which gives the M_p diagram at (e) below which are tabulated values of M_p M_p/I. At (f) is the deflection diagram for the auxiliary 1-lb. load at D, and at (g) the values of M_p M_p/I are shown

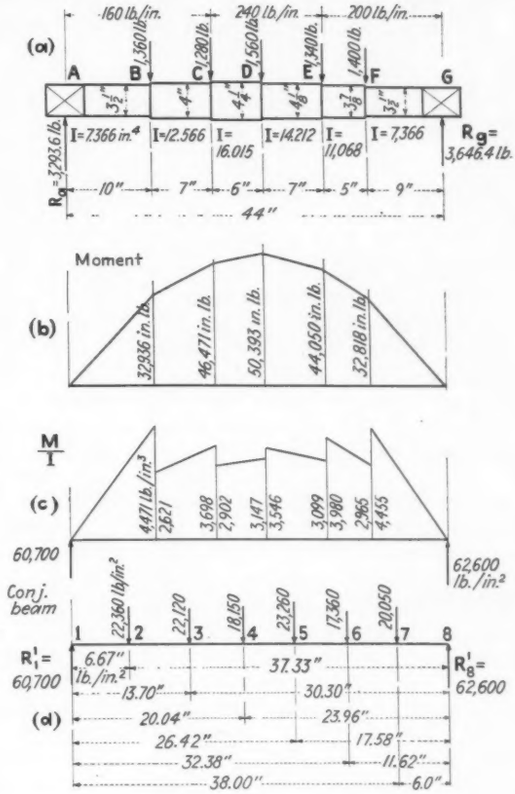


Fig. 3—The M/I diagram method for determining critical speed of a loaded shaft

Now applying Equation (1),

$$f = \frac{1}{2\pi}\sqrt{\frac{g\Sigma W_i y_i}{\Sigma W_i y_i^2}} = \frac{1}{2\pi}\sqrt{\frac{g\Sigma y_i}{\Sigma y_i^2}}$$

$$\Sigma y_i = 0.0929 + 0.1034 + 0.0929$$

$$\Sigma y_i^2 = \overline{0.0929}^2 + \overline{0.1034}^2 + \overline{0.0929}^2$$

$g = 32.2 \times 12 = 386$ in./sec./sec.

$f = 10.06$ cycles/sec.

The critical r.p.m. is found to be:

$(\text{r.p.m.})_{cr} = 60 \times 10.06 = 603$ r.p.m.

If the shaft consists of many short sections of different diameters carrying numerous loads, the foregoing method becomes quite tedious since the auxiliary load must be applied at every point where the deflection is required.

For such a shaft the conjugate beam ("Elements of Strength of Materials," Timoshenko and MacCullough, Second Edition, 1940, p. 167, D. Van Nostrand Co.) is suitable. In this process a fictitious or conjugate beam is assumed to be carrying a load distributed similarly to the *M/I* diagram of the given or actual beam. The bending moment which occurs at any point in this conjugate beam, when divided by *E*, will be the deflection of the actual beam at that point.

For a shaft having the uniform loads shown in Fig. 3(a) the *M/I* diagram will be simpler and much labor will be saved if the load on each interval will be considered, not as concentrated at its mid-point, but as divided into two equal parts located at the end points of the segment. Such a redistribution of the load will give the moment diagram of Fig. 3(b), and at (c) the *M/I* diagram.

While Fig. 3(c) is the conjugate beam for this shaft, it is usually satisfactory to assume that all of the load of each interval is concentrated at the center of gravity of the triangles and trapazoids of (c). In Fig. 3(d), these loads and their respective locations, together with the reactions, which are found in the usual way, are shown.

The deflection at 2 will then be:

$$y_2 = \frac{60{,}700 \times 6.67}{30{,}000{,}000} = 0.0135 \text{ in.}$$

At 3, the deflection is found to be:

$$y_3 = \frac{60{,}700 \times 13.70 - 22{,}360 \times 7.03}{30{,}000{,}000} = 0.0225 \text{ in.}$$

The deflection for the other points are found in a similar manner.

Point	Load	y	Wy	Wy^2
2	1,600	0.0135	21.60	0.2916
3	1,120	0.0225	25.20	0.5670
4	1,440	0.0259	37.30	0.9660
5	1,680	0.0255	42.84	1.0924
6	1,000	0.0205	20.50	0.4203
7	1,800	0.0125	22.50	0.2812
			169.94	3.6185

Applying Equation (1)

$$f = \frac{1}{2\pi}\sqrt{\frac{386 \times 169.94}{3.6185}} = 21.43 \text{ cycles/sec.}$$

The critical r.p.m. is found to be,

$(\text{r.p.m.})_{cr} = 60 \times 21.43 = 1{,}286$ r.p.m.

A slight improvement in this result can be obtained by considering each of the foregoing loads as located at the center of its interval. Corresponding value of y is found by interpolation of the values above. The actual bending moment diagram as produced by the given loading should always be drawn for such problems. Inspection will then usually serve to determine whether the diagram given by the rearranged loading is sufficiently close to the true value to yield accurate results.

FATIGUE STRENGTH—III

Of Members as Influenced by Surface Conditions

O. J. HORGER, *Director, Railway Engineering and Research*
The Timken Roller Bearing Company

INVESTIGATIONS covering the influence on fatigue endurance of operations such as burnishing, rolling and press-fitting, and also the effects of surface conditions on fatigue of members under torsional loading were described in Part II, which appeared in the December 1940 number of PRODUCT ENGINEERING. This concluding installment will deal with the fatigue strength of low temperature quenching and surface treatments.

Forging operations under some conditions may produce directional properties because of grain flow. Static ultimate and yield strength properties show little or no difference from specimens selected with or against grain flow. Ductility and impact properties, however, of transverse specimens are lower than for those selected parallel to the grain flow. Investigators have reported in some instances large variations and in others little change in fatigue strength for specimens taken along and across the grain flow.

Aitchison and Johnson when investigating the directional properties of various steel forgings (See "The Effect of Grain Upon the Fatigue Strength of Steels," *Journal of the Iron and Steel Institute*, Vol. CXI, No. 1, 1925, p. 351–378) found that the fatigue strength of specimens selected parallel and across the grain flow showed a maximum difference in some instances of not quite 20 per cent. Also Hauttman when testing selected specimens parallel and across the grain from crank sections of large marine engine crankshafts (See "The Relation of Laboratory Research to Engine Design," *Trans. N. E. Coast Institute of Engineers and Shipbuilders*, January 1936) found the fatigue strength of specimens across the grain to be 25 per cent less than that parallel with the grain. H. Wiegand (See "The Internal Effect of Notches and Fatigue Strength," *Metallwirtschaft*, Vol. 18, Jan. 27, 1939, p. 83–85) investigated quenched and tempered steel forgings (0.30 per cent C; 2.0 per cent Cr.; 1.5 per cent Ni.) from two different sources of supply. Table IV shows results of the investigation.

Fatigue failures have been reported in railroad axles and forgings, in which were found outcroppings of flow lines to the surface. Forgings sometimes include laps, or hammered-in scale, or are marked in upsetting, all of which are points of weakness under variable stresses. In Fig. 15 is shown a fatigue failure that developed at the end of an upset mark forming a small shoulder. Failures did not occur in similar members not containing this condition although no machining was done.

Forgings which are not machined exhibit a lower fatigue strength as compared with forgings having machined surfaces. Fig. 16 shows diagrammatically the results of fatigue studies on various steels as made by G. A. Hankins and M. L. Becker (See "The Fatigue Resistance of Unmachined Forged Steels," *Journal Iron and Steel Institute*, Sept. 1932). Note that the effect of the forging skin becomes greater with increase in tensile strength. Also there is considerable discrepancy in some instances between the values for the same material obtained from the two manufacturers.

Structural conditions are sometimes found similar to those indicated in Fig. 17*(A)* which shows the macroetch of a transverse section of a failed 7⅝-in. dia. machined forging of S.A.E. 1045 steel. Plane of the fatigue fracture is given in Fig. 17*(B)* with the ingot pattern from *(A)* superimposed and properly oriented as indicated by the broken line. Fatigue failure developed in the region of the corner of the pattern. It is not known what weakening effect, if any, such conditions have on the fatigue strength but it has been observed that some fatigue failures definitely initiate from the region of the corner of the pattern. Patterns such as the one shown which lie in the boundary regions of the forging are generally only found in steel taken from the lower portion of ingots.

Low Temperature Quenching

Fatigue properties of materials which have been subjected to conventional heat-treatments are common knowledge, however, little practical use has been made of the beneficial results obtained by low temperature quenching. Several foreign investigations have been made where quenching was done at temperatures generally used for tempering. Tensile test values are changed little by this treatment from the values obtained by the ordinary treatment used to obtain similar static properties but the fatigue limit may be considerably increased. This increase in fatigue resistance is explained by the presence of residual compressive stresses developed in the surface by such low temperature quenching. It has been well established that such initial surface compressive stresses increase the fatigue strength and that tensile stresses cause a reduction in fatigue strength. In view of the importance of low temperature quenching some pertinent results are of practical importance.

Comprehensive investigation of quenching below the lower critical temperature was initiated by H. Bühler and H. Buchholtz in which they correlated the fatigue properties with the magnitude of the residual stresses present as well as with the usual tensile test values (see "Effect of Residual Stresses on the Bending Fatigue Strength," *Stahl und Eisen*, Vol. 53, 1933, p. 1,330). Their conclusions are shown diagrammatically

Fig. 15—Large shaft with fatigue failure originating at upset mark

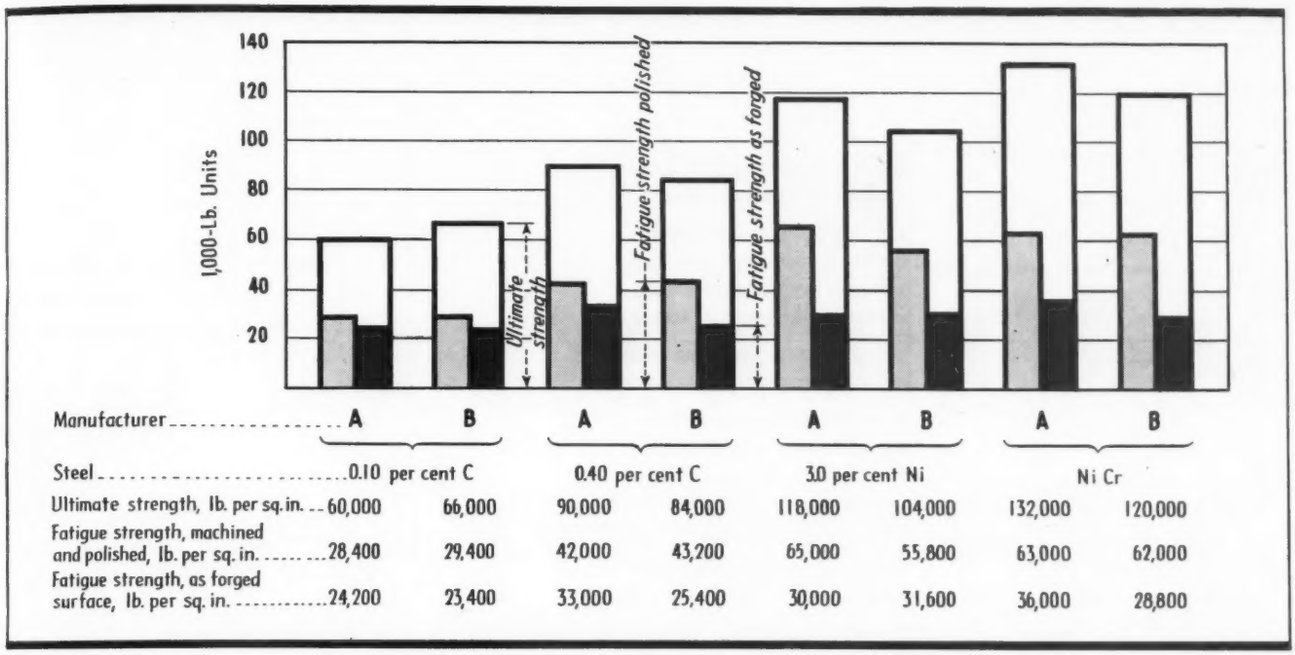

1,000-Lb. Units
140
120
100
80
60
40
20
Ultimate strength
Fatigue strength polished
Fatigue strength as forged
Manufacturer ..... A B A B A B A B
Steel ..... 0.10 per cent C 0.40 per cent C 3.0 per cent Ni Ni Cr
Ultimate strength, lb. per sq. in. .. 60,000 66,000 90,000 84,000 118,000 104,000 132,000 120,000
Fatigue strength, machined and polished, lb. per sq. in. 28,400 29,400 42,000 43,200 65,000 55,800 63,000 62,000
Fatigue strength, as forged surface, lb. per sq. in. 24,200 23,400 33,000 25,400 30,000 31,600 36,000 28,800

Fig. 16—Effects of machined and as-forged surfaces on endurance fatigue strength of steel, investigated by Hankins and Becker

in Fig. 18 where increases of as much as 20 per cent are indicated.

Bending fatigue tests on rolled flat plates used for spring leaves, 18x1¼ x⅜ in., of chromium-vanadium and low chromium steels, have been made by M. L. Becker and C. E. Phillips (See "Internal Stresses and Their Effect on the Fatigue Resistance of Spring Steels," *Journal of Iron and Steel Institute*, Vol. 133, Number 1, 1936). Their results for unmachined plates showed that the range of allowable fatigue stress was increased by as much as 36 per cent by quenching in water from a suitable tempering temperature instead of employing the usual oil quench and temper.

Tangible evidence of the practical application of Becker and Phillips' laboratory tests is submitted by one firm. They state that carbon manganese shafts, which were being given a final heating at 1,110 deg. F. and cooled in

Table IV — Bending Endurance Limit of Steel Forgings

Specimen Location	Lb. per sq. in.	
	Source A	Source B
Parallel grain flow	53,000	60,000
Transverse to grain flow	50,000	44,000

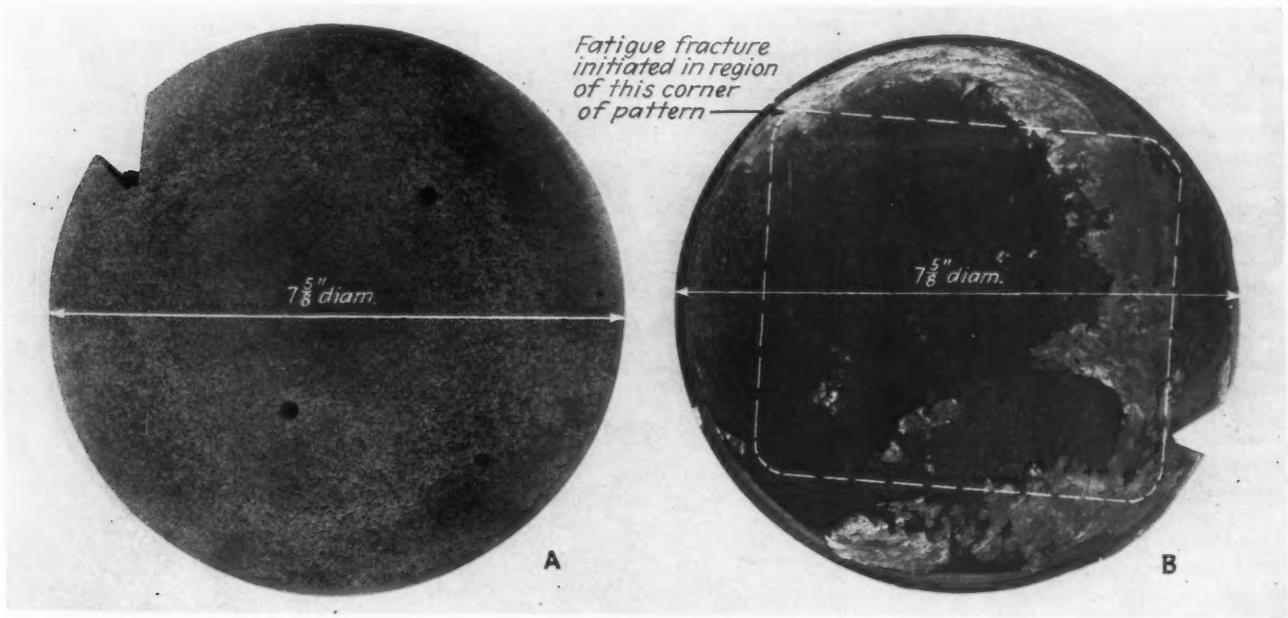
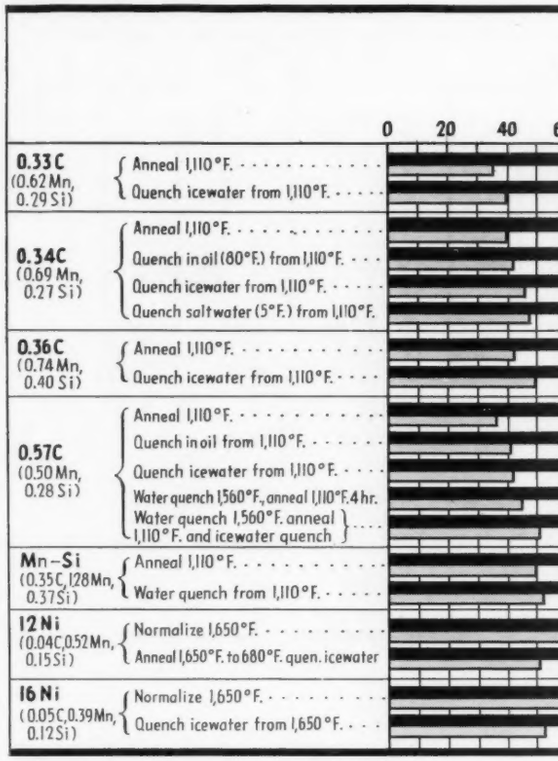


Fig. 17—Correlation of structural pattern with source of fatigue failure. Holes were drilled and V-shaped piece cut out after failure. (A) Macroetch showing structural pattern. (B) Pattern from (A) superimposed on fatigue fracture



		YIELD STRENGTH LB. PER SQ. IN.	ULTIMATE STRENGTH LB. PER SQ. IN.	ELONGATION PER CENT	BENDING FATIGUE LIMIT, LB. PER SQ. IN.	INTERNAL STRESS IN SURFACE, LB. PER SQ. IN. +=TENSION −=COMP.	
						LONGITUDINAL	TANGENTIAL
0.33C (0.62 Mn, 0.29 Si)	Anneal 1,110°F.	55,700	84,700	53	35,500	0	0
	Quench icewater from 1,110°F.	54,800	89,300	51	39,800	− 32,700	− 39,800
0.34C (0.69 Mn, 0.27 Si)	Anneal 1,110°F.	49,600	91,500	54	39,800	0	0
	Quench in oil (80°F.) from 1,110°F.	54,900	94,700	54	42,700	− 29,200	− 29,200
	Quench icewater from 1,110°F.	51,700	96,600	54	46,900	− 45,500	− 45,500
	Quench saltwater (5°F.) from 1,110°F.	50,200	93,300	55	48,300	− 48,300	− 45,500
0.36C (0.74 Mn, 0.40 Si)	Anneal 1,110°F.	52,200	98,800	−	42,700	0	0
	Quench icewater from 1,110°F.	52,700	98,000	−	49,800	− 45,500	− 39,800
0.57C (0.50 Mn, 0.28 Si)	Anneal 1,110°F.	45,500	90,900	35	37,000	0	0
	Quench in oil from 1,110°F.	44,100	92,500	35	41,300	− 34,100	− 28,500
	Quench icewater from 1,110°F.	44,100	95,700	32	42,700	− 48,300	− 42,700
	Water quench 1,560°F., anneal 1,110°F. 4 hr.	86,800	115,700	60	45,500	0	0
	Water quench 1,560°F. anneal 1,110°F. and icewater quench	86,800	118,000	60	51,200	− 52,700	− 45,500
Mn−Si (0.35C, 1.28 Mn, 0.37 Si)	Anneal 1,110°F.	62,200	99,500	58	49,800	0	0
	Water quench from 1,110°F.	61,400	101,000	58	52,700	− 24,200	—
12 Ni (0.04C, 0.52 Mn, 0.15 Si)	Normalize 1,650°F.	115,000	149,300	49	58,200	0	0
	Anneal 1,650°F. to 680°F. quen. icewater	113,700	152,500	41	51,200	+ 52,700	+102,300
16 Ni (0.05C, 0.39 Mn, 0.12 Si)	Normalize 1,650°F.	121,500	167,000	40	62,500	0	0
	Quench icewater from 1,650°F.	127,000	166,000	44	52,700	+ 18,500	+42,700

Fig. 18—Effect of low temperature quench on fatigue strength from investigations of Bühler and Buchholtz

air, were developing fatigue failures. These shafts when water quenched from 1,110 deg. F. gave no trouble.

Surface Treatments

Case carburizing and nitriding lead to large increases in fatigue strength, the advantages of such treatments are well known. Maximum fatigue strength is not obtained with such hard surfaces, however, under conditions of friction and accompanying temperatures. An example of this is indicated in Fig. 19 which shows a fatigue failure of a case carburized member subjected to alternate bending stresses. The stresses alone were not large enough to cause failure but in this instance a frictional load was transmitted by a bronze ring operating intermittently on the member, fatigue failure initiated immediately under the ring. Also applications of chrome and nickel platings to highly stressed members are not usually recommended.

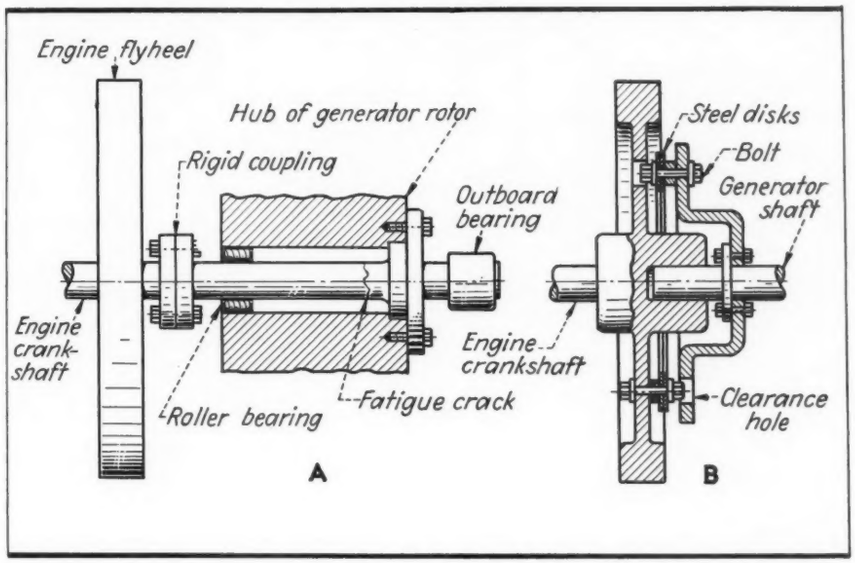
Induction and flame hardening methods are meeting with very wide use, principally because of the ability to increase wear resistance in local regions. In addition, these methods also offer great increases in fatigue strength although too little advantage is taken of this in practice. Flame hardened fillets showed 80 per cent increase in endurance fatigue limit. Shafts having press-fitted wheels gave over 200 per cent increase as indicated in Table III (p. 564, Dec. 1940 *P.E.*). Fatigue tests on flame hardened 7½-in. members indicate that appreciably increased strength may be expected.

Metal spraying of shafts has been investigated to determine its influence on the fatigue strength. Because it is often necessary to build up worn shafts, rotating bending fatigue tests were made on 1¾-in.-dia. shafts of 0.35 per cent carbon steel which were metal sprayed with a ⅛-in. thick layer. These built-up shafts being 2 in. in dia. showed practically the same endurance fatigue limit as 2 in. shafts not metal sprayed. Also in press-fitted members, the fit is often lost at times of disassembly so that it would be convenient to restore the fit by building up the axle. For this reason, fatigue tests were made on 2 in. axles which were metal sprayed in the fitted portion and wheels pressed-on. Results, as indicated in Table III (p. 564, Dec. 1940 *P.E.*), showed that the metal-sprayed shaft had 62 per cent greater fatigue than the axles not sprayed.

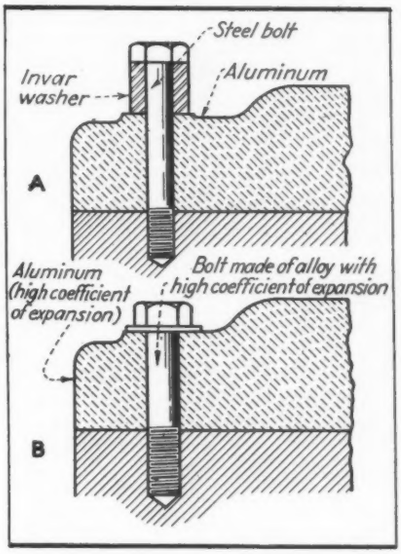
Fig. 19—Bending fatigue failure of case carburized member which initiated in region of applied frictional forces

CAUSES AND CURES

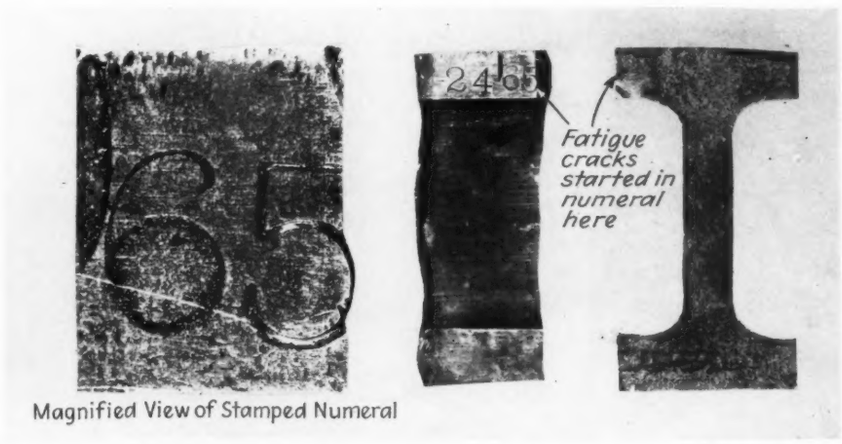
PRODUCT ENGINEERING will pay $3 for each example published in Causes and Cures. Where illustrations are necessary, include drawings, rough sketches or photographs



In close-coupling a generator to a diesel engine, the arrangement shown schematically in diagram *A* was designed to absorb torsional vibration of the engine. Although the flexibility of the long shaft absorbed considerable vibration, fatigue cracks started at the foot of the fillet near the end of the shaft, as shown and resulted in failure of the shaft. Cure was the substitution of the coupling shown in *B*. Torsional flexibility is provided by the steel disks.



Aluminum parts bolted to steel parts may create considerable difficulties because of the high coefficient of expansion of aluminum. For example, steel bolts or studs through aluminum engine parts are likely to be overstressed when the engine runs hot. To overcome this difficulty, the English have developed a steel alloy for bolts, known as N.M.C., that has a high coefficient of expansion. Another method is to use an extra long steel bolt and place a thick washer of low-expansion Invar under the head.



Magnified View of Stamped Numeral

Numerals stamped in a high-stress section of this forged machine part caused this fatigue failure. Cracks started in the curved portion of the numeral 5, as shown. Note that this curve of the numeral, and similarly for the numeral 6, is stamped deeper than the rest of the numerals, probably because the stamp was tipped when struck with the hammer. Although polished forged steel may show a fatigue resistance value of 41,000 lb. per sq.in., a study of failures such as this shows that the presence of tool marks or scratches may reduce this value to as low as 19,000 lb. per sq.in. Recurrence of this failure was prevented by substituting another method of marking.

DIFFICULTY IN WELDING machine bases was encountered because of distortion. It was found that this could be reduced by flame-cutting openings and lightening holes in certain parts of the plate after the base had been fabricated by welding. Another solution was to weld in tubular braces which were not removed until after the welded structure had been stress-relieved.

ALLOY STEEL PLATES often crack in forming and shearing, and distort when welded or flame-cut. This trouble is usually caused by internal stresses set up in the rolling operation. Cure is to work the plates in normalized condition, achieved after rolling by heating them to 1,600–1,800 deg. F. and cooling in air.

PERMANENT MAGNETS IN

Permanent magnets cast or sintered from high-coercive aluminum-nickel alloys often are the simplest and most effective means for performing mechanical operations. These illustrations show how they can be designed to replace springs, clamps, over-center mechanisms, to transmit motion, and to translate speed into torque. Several unique methods of mounting these hard-to-machine magnets are shown in addition to the common methods of welding, brazing, soldering, inserting in die-castings, and fastening by rivets, screws, through-bolts and clamps. See also "Permanent Magnets in Electrical Applications," PRODUCT ENGINEERING for December, 1940.

Fig. 1—Theft-proof electric light socket, of which mechanical principles only are shown, uses permanent magnet "key" which is cast in two halves pinned and magnetized so that one claw is north pole, the other south pole.

Fig. 2—A small cast permanent magnet placed in the cover of the die-cast raceway of this conduit system permits quick and easy location of outlet many years later. A compass-type finder does the trick.

Fig. 3—Magnets of aluminum-nickel alloy perform holding operation in this snap-action limit switch of General Electric Company. As operating dog shifts, spring tension builds up tending to shift yoke assembly which is held stationary by the magnet. When operating dog approaches end of travel, direct mechanical contact breaks yoke away from holding magnet. Spring then snaps yoke to opposite side.

Fig. 4—Unique application of small permanent magnets is in plastic letters of "Quixet" movie titler of Hamilton Dwight Company. Magnets hold letters in place on steel title sheet.

Fig. 5—Beauty shops find a use for permanent magnets in attractive hairpin pullers such as shown here. Magnet is inserted into hot Catalin case; shrinkage of plastic on cooling grips unit firmly.

Fig. 6—Two aluminum-nickel cast magnets are used in this Frantz FerroFilter for removing iron particles from lubricating systems. A soft lead core is cast in one magnet over which is flared a brass tube, and the other magnet is fastened by pressing on a brass band.

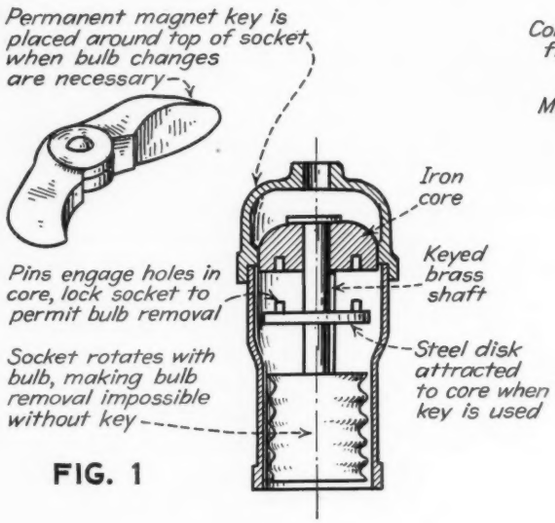

Permanent magnet key is placed around top of socket when bulb changes are necessary
Iron core
Keyed brass shaft
Pins engage holes in core, lock socket to permit bulb removal
Steel disk attracted to core when key is used
Socket rotates with bulb, making bulb removal impossible without key

FIG. 1

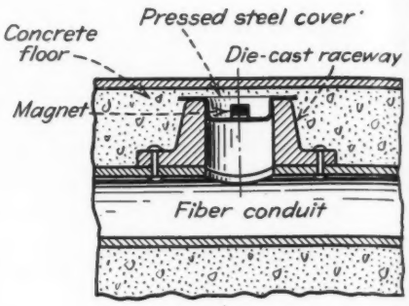


FIG. 2

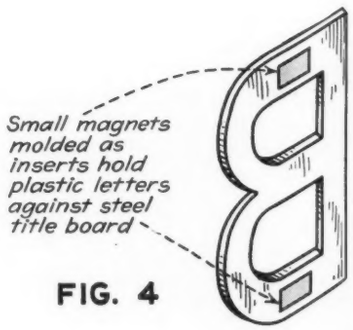


FIG. 4

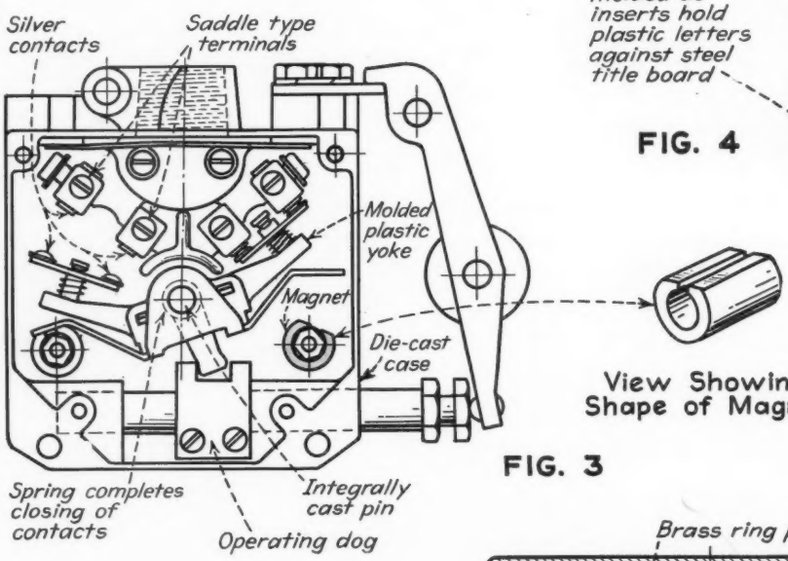


FIG. 3

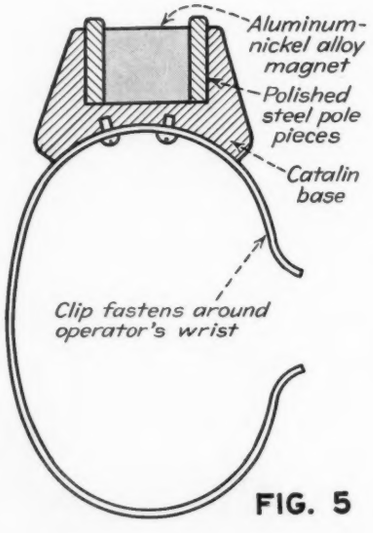


FIG. 5

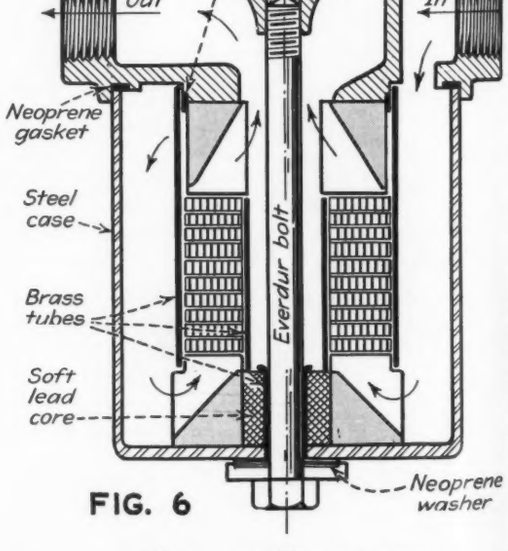


FIG. 6

MECHANICAL APPLICATIONS

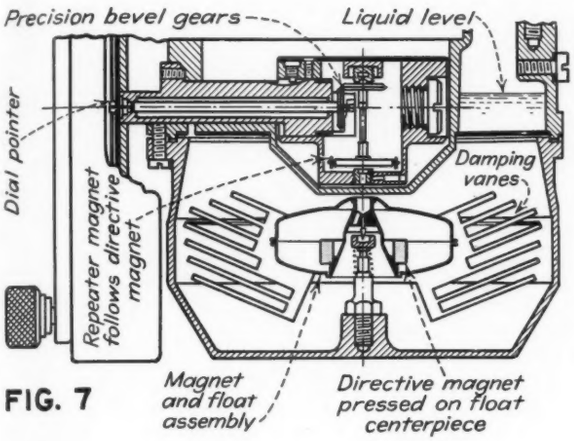


FIG. 7

Fig. 7—Sintered magnet in the float of this Kollsman direction indicator transmits its rotary motion to the cobalt-steel follower magnet located on the other side of the aluminum wall of the float case, and thence through bevel gears to the dial pointer shaft. Magnetic couplings of this type are often used to transmit motion through solid partitions.

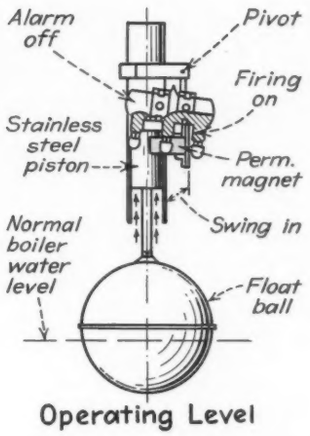


FIG. 8

Fig. 8—Magnet cast of aluminum-nickel alloy in this Schaub Magnetrol operates mercury switch to control boiler water level. For applications where the float is free to slide up and down the rod between limits, a magnetic clutch (not shown) holds rod and cylinder in the up position until the float reaches lower limit stop.

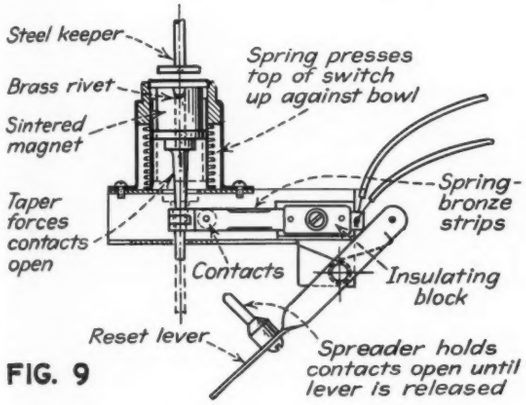
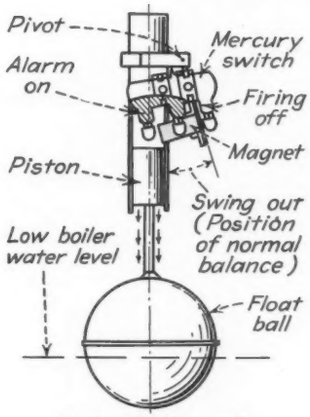


FIG. 9

Fig. 9—Magnetic switch on General Electric glass coffee maker automatically shuts off heating element when upper bowl fills. With current "on" magnet is held up by a steel keeper mounted at the end of a rod in upper bowl stem. When the last cup or so of water rises through the stem, turbulence against a disk on upper end of keeper rod lifts keeper from bottom of bowl, releasing magnet which opens contacts.



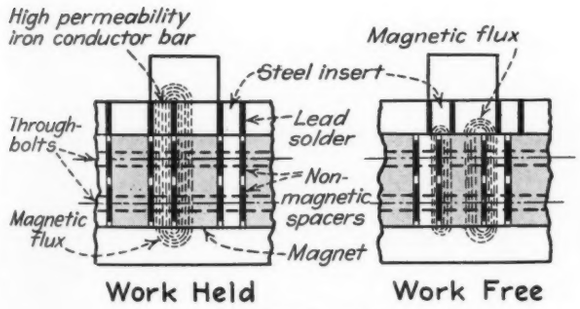


FIG. 10

Fig. 10—Brown & Sharpe magnetic chucks illustrate design of shunts to turn permanent magnet holding devices on and off. Chuck consists of a pack of rectangular-shaped aluminum-nickel magnets and soft steel conductor bars separated by non-magnetic spacers. Pack slides between face plate and bottom plate by means of crank and eccentric. Positions of pack when "on" and "off" and paths of magnetic flux are shown in the diagram. All joints between component parts are ground to close fits to reduce reluctance.

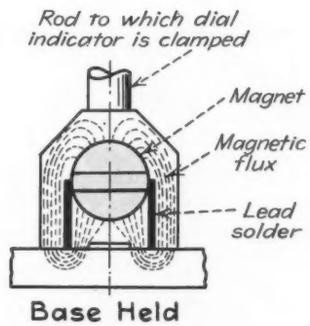
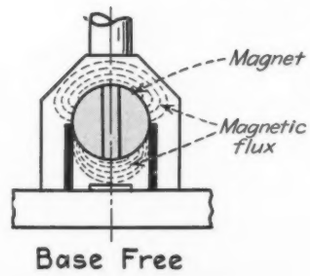


FIG. 12

Fig. 12—Magnetic dial indicator base can be turned on and off by 90-deg. rotation of hand knob on end of cylindrical magnet. Principle of operation is similar to Fig. 10.



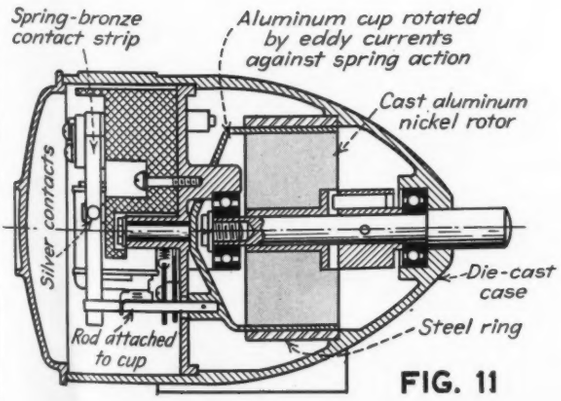


FIG. 11

Fig. 11—Plugging relay of General Electric Company is operated by eddy currents set up in aluminum cup by rotor of aluminum-nickel magnet alloy. Magnetic reaction produced turns the cup through its limited rotation, closing contacts against spring action. As rotor slows down, a definite point is reached where spring force overcomes magnetic force, at which point the contacts trip open.

STEEL TUBE FUSELAGES – IV

Designs of Fittings

JAMES E. THOMPSON and T. A. LOWE
Vultee Aircraft, Inc.

AFTER completing the basic tube structure design and deciding upon the exact nature and location of the major fittings required for engine, wing and empennage attachments, the sole remaining problem is the determination of detail fitting design. Detail fittings can be grouped into the following classifications: (1) instrument panel, (2) air controls, (3) engine controls, (4) pilot's seat, (5) communication and electrical equipment, (6) armament, (7) furnishings.

Two simple factors govern the basic design of all detail fittings, namely, welding accessibility and balanced joints. The wall thicknesses and sheet gages of the major fittings are usually large enough to minimize balanced joint considerations. But most detail fittings are thin sheet metal brackets, clips, or lugs welded to comparatively heavy wall tubing, or vice versa. To further complicate their design, the space available for many of these detail fittings is limited and considerable ingenuity is required to develop brackets which will permit adequate clearances for rapid production with standard welding equipment.

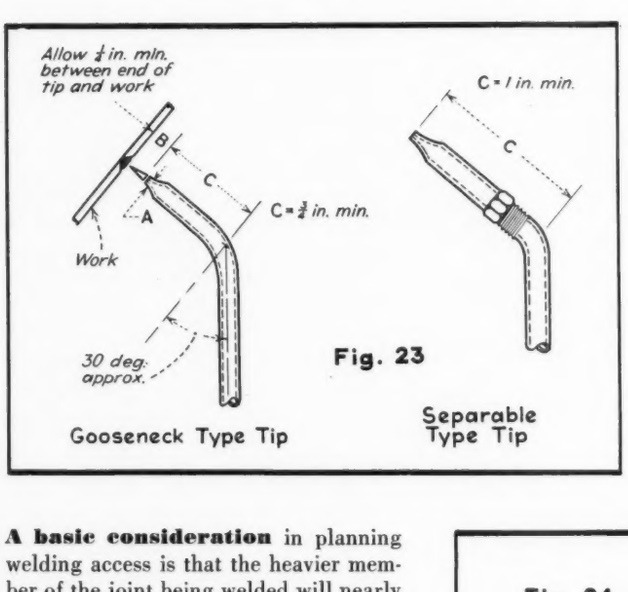


Fig. 23

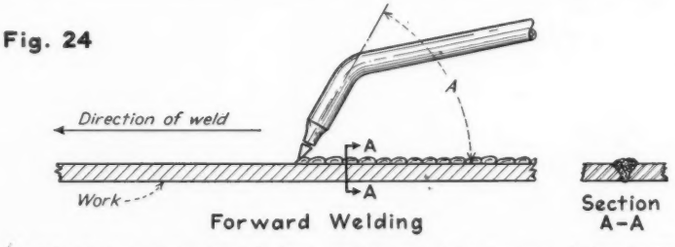
Welding Clearance

In the design of every welded joint adequate access as determined by the size of the welding equipment used, must be provided for application of the torch flame or arc electrode to every point requiring fusion. General dimensions of a typical welding torch tip are shown in Fig. 23. Orifice diameter *A*, the length of the neutral cone *B*, and the length *C* of the gooseneck bend will increase with tip size. For working in close quarters, dimension *C* can be reduced to about ⅜ in. by cutting off a tip at the bend, telescoping a short length of copper tubing into the remaining stub, silver-soldering it in place and swaging down the open end to the desired orifice. However, such a tip should rarely be considered for production. All joints requiring welding in extremely limited space should be checked with an experienced welder to determine whether joint to be welded is adequately accessible.

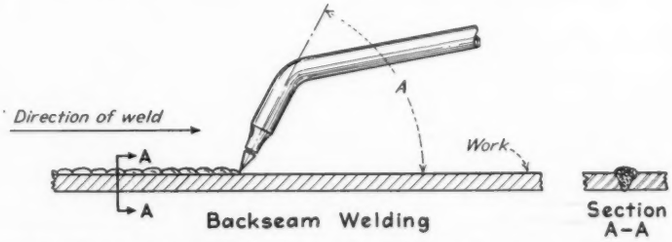
A basic consideration in planning welding access is that the heavier member of the joint being welded will nearly always require the greater heat, and hence it will be necessary to bring the welding flame more directly against this member, rather than against the lighter member. Fig. 24 shows examples of some of the basic factors governing application of the welding torch. A study of these will enable one to understand better why the conditions shown in the sketches are "correct" or "incorrect" from a practical production viewpoint.

The ideal joint from a production standpoint is one wherein the cross-sectional area in both pieces being joined is balanced or equal. It is then unnecessary for the welder to exercise particular care in the application of the flame in order to avoid burning the thin section before the thick section has been brought to the proper welding temperature.

Fig. 24



Angle "A" will be approximately 60 deg. in most cases, but may vary between 30 deg. and 90 deg., depending upon the thickness of the pieces being welded.



Angle "A" will vary between 45 deg. and 90 deg. This type of welding can be depended upon to obtain extremely good penetration to the bottom of the joint "vee".

Balanced Joints

An ideally balanced joint, both pieces of equal cross-sectional area, is shown in Sketch I of Fig. 25. Welding of badly unbalanced joints is possible but slow; therefore joints should be as well balanced as practicable.

In the unbalanced joint shown in Sketch II of Fig. 25, greater welding time will be required. The torch must be held at a fairly close angle so as to apply most heat to the thicker part, and welding along the edge of the thinner sheet will be slow.

Thin gusset welded normal to a heavy base member, Sketch III of Fig. 25, illustrates a second type of unbalanced joint; the edge of the thin gusset will heat more rapidly than the flat base plate. The thickness X of the base plate should never be more than three times T. Ideally, T should be slightly greater than X.

Sketch IV of Fig. 25 shows a badly unbalanced joint. Depth D should not exceed the gap W if W is less than ¾ in. or if the weld seams are 1½ in. or more in length. If web T is comparatively thin in relation to plate thickness X the gap must be large enough to permit directing the greater part of the welding heat against the heavy plate members.

The general principles of balanced joints apply to all gas or arc welded sheet-to-sheet, sheet-to-tube, or tube-to-tube joints. Arc electrodes can be bent through a considerable angle without cracking the protective flux coating, in order to obtain access for welding in limited space.

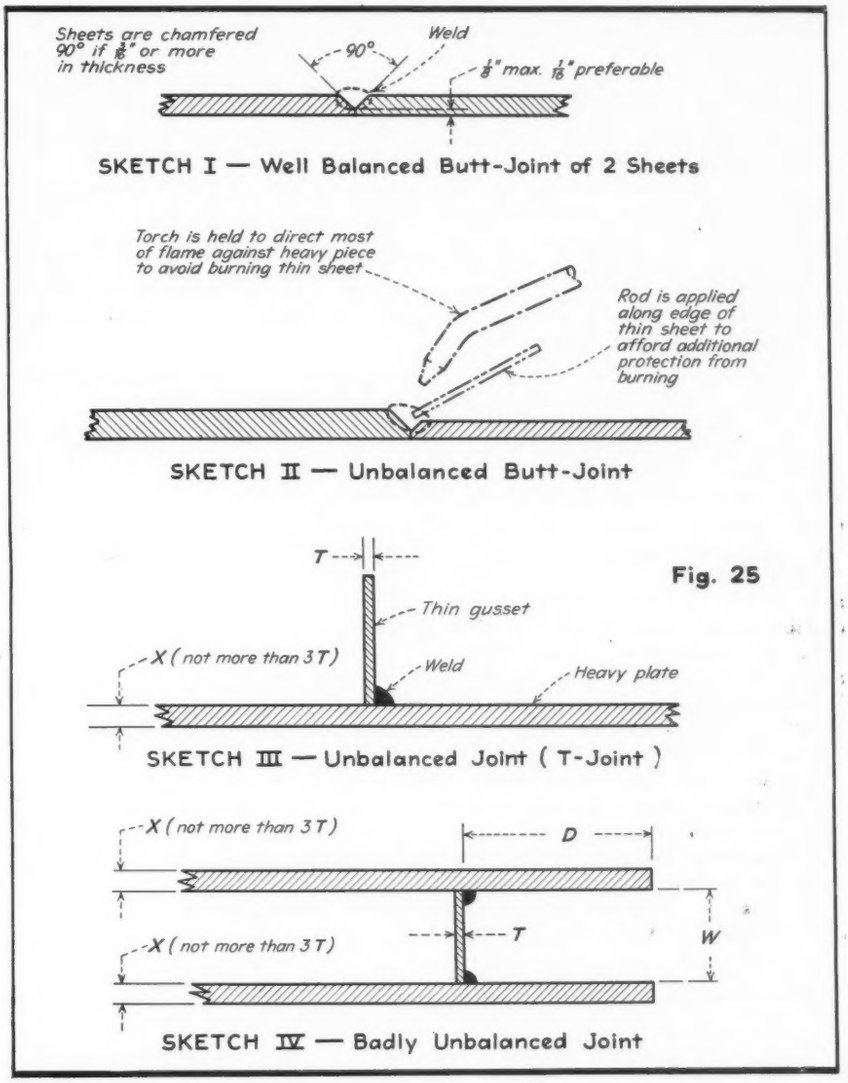

Sheets are chamfered 90° if ⅛" or more in thickness
90°
Weld
⅛" max. 1/16" preferable
SKETCH I — Well Balanced Butt-Joint of 2 Sheets
Torch is held to direct most of flame against heavy piece to avoid burning thin sheet
Rod is applied along edge of thin sheet to afford additional protection from burning
SKETCH II — Unbalanced Butt-Joint
T
Thin gusset
X (not more than 3 T)
Weld
Heavy plate
SKETCH III — Unbalanced Joint (T-Joint)
X (not more than 3 T)
D
T
W
X (not more than 3 T)
SKETCH IV — Badly Unbalanced Joint

Fig. 25

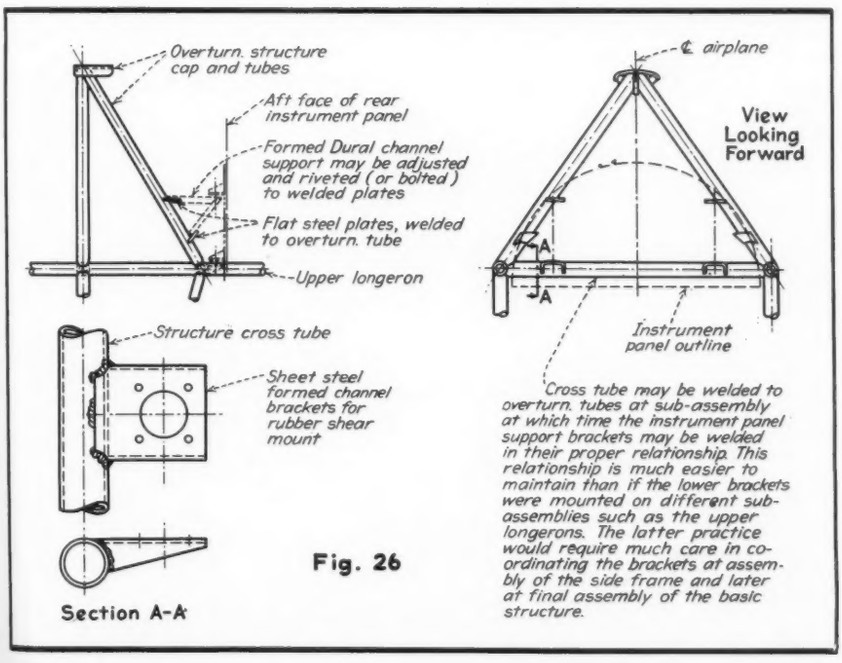

Overturn. structure cap and tubes
Aft face of rear instrument panel
Formed Dural channel support may be adjusted and riveted (or bolted) to welded plates
Flat steel plates, welded to overturn. tube
Upper longeron
Structure cross tube
Sheet steel formed channel brackets for rubber shear mount
Section A-A
℄ airplane
View Looking Forward
A
A
Instrument panel outline
Cross tube may be welded to overturn. tubes at sub-assembly at which time the instrument panel support brackets may be welded in their proper relationship. This relationship is much easier to maintain than if the lower brackets were mounted on different sub-assemblies such as the upper longerons. The latter practice would require much care in co-ordinating the brackets at assembly of the side frame and later at final assembly of the basic structure.

Fig. 26

Instrument Panels

Instrument panel assemblies weigh as much as 40 lb. for a military panel with all engine and flight instruments, and in size are often equivalent to a 24-in. square, or even larger. Panels must be securely supported in anti-vibration mount bushings, usually of the rubber "shear" type.

The conventional instrument panel mounting is a four-point suspension with the attachment points near the corners. When mounted in a steel tube fuselage the panel base is usually supported by fittings welded to the main fuselage structure, usually to a cross tube of that structure, and the upper portion of the panel is connected to a truss frame rising from the main structure forward of the panel. In two-place tandem-cockpit airplanes, it is frequently possible to dispense with the truss frame for supporting the upper

portion of rear cockpit instrument panel, and, as shown in Fig. 26, utilize the vertical tubes of the overturn structure for this purpose.

Sometimes for small light panels, but not for large heavy panels, the panel mount bushing upper supports are attached to stiffened portions of the cockpit enclosure deck structure immediately forward of the cockpit, and the lower supports are attached to trim panels used to fair the instrument panel to the fuselage sides. The deck and trim panels are attached to the basic fuselage structure by welded lugs.

Attachment points should be related as closely as possible and must be planned to avoid complications arising from accumulation of manufacturing tolerances on the various non-related parts to which the panel supports connect. If one of the attachment fittings is a welded sub-assembly, such as a tube truss supporting the top of an instrument panel, installation drawing of this sub-assembly in the basic structure should be dimensioned to show the positions of its attachment lugs or bosses with respect to related attachment points on the basic structure.

The dimensional limits for related parts of a welded structure should not be closer than ± 1/32 in., even for parts within the same sub-assembly. When closer tolerances are required, such as center distances between mounting holes, allow sufficient stock for shifting the hole boss or lug during welding and drill the holes after welding.

Air Controls

Basic air controls usually include a control stick or column assembly and a pair of rudder pedals in each cockpit, attached to cables passing over suitable deflecting pulleys to connect with the control surfaces, that is, ailerons, elevators and rudder. Tandem-cockpit airplanes provided with air controls in both cockpits can use either two identical assemblies, interconnected between the cockpits by cables, as shown at Fig. 27, or one unit assembly comprising two stick assemblies joined by an aileron torque tube. The former should be planned for installation to the fuselage structure, while the latter can frequently be installed to best advantage by bolting directly to brackets attached to the main wing beams. The suspended stirrup-type rudder pedal is commonly used and lends itself to simple installation as shown at Figs. 28 and 29. A cross tube is welded between suitable fuselage side-tubes and located to correctly position the pedals forward of the pilot's seat, each pedal being supported by two pairs of lugs welded to the lower side of the cross tube.

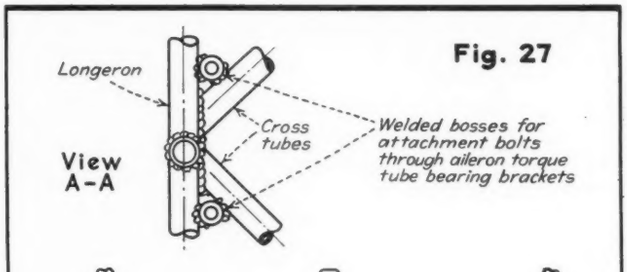


Fig. 27

CORRECT

If a series of lugs are to be welded and line drilled it is advisable to make R_2 large enough to allow for weld distortion and yet maintain sufficient edge distance at hole. In most cases the distortion will not exceed ±1/16".

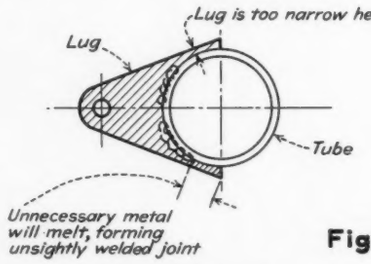


Fig. 28

INCORRECT

Even if the welder avoided burning the narrow strip away, the value of the remaining base metal will not justify the care and skill required.

CORRECT

*Tube, end plate and lugs are welded and drilled as one sub-assembly. The dimension **A** is greater than **B**, allowing sufficient length of plate to weld to the structure and also sufficient clearance for line-drilling after sub-assembly. Dimension **D** should be at least 1/16" to allow sealing the cross-tube by welding entirely around the tube at its junction with end-plate.*

This design can be improved by fitting the cross-tube on the centerline of the structure tube, and thus entirely eliminating the end-plate.

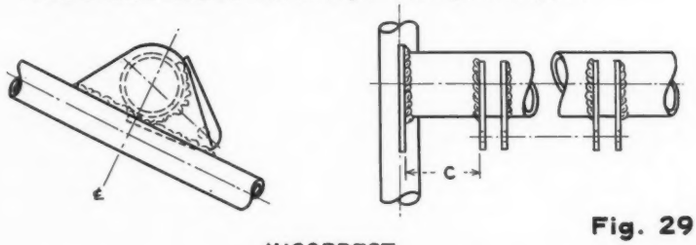
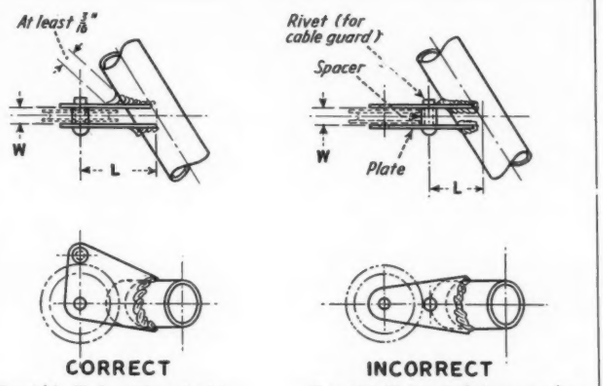


Fig. 29

INCORRECT

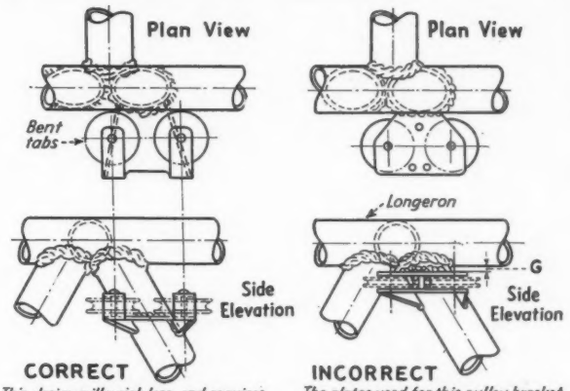
*End-plate interferes with line-drilling of lugs, necessitating that this be done before plate is welded to tube, and necessitating an additional sub-assembly operation and jig. The second sub-assembly operation of welding plate to tube can be eliminated only if the dimension **C** is sufficient to permit using an offset drill spindle, and this should not be considered for a production design.*



The width W is not important in this design as all welding is done on the outside surfaces of the plates. The length L is also unimportant, as the rivet is at one side of the bracket, well away from the weld beads.

The width W is entirely to narrow for weld access, and at the best it would only be possible to weld one plate on both sides, and then weld the second plate on the outside. The length L is too short, causing interference between rivet and weld bead.

Fig. 30



This design will weigh less, and requires only about half the amount of welding. The fabrication of the plate can be further simplified in the case of lightly stressed pulleys by spotwelding washers on the underside of the plate to provide adequate bearing area for cantilever axle bolts. This eliminates the bent-up tabs, and rivets and spacers are used to form cable guard pins.

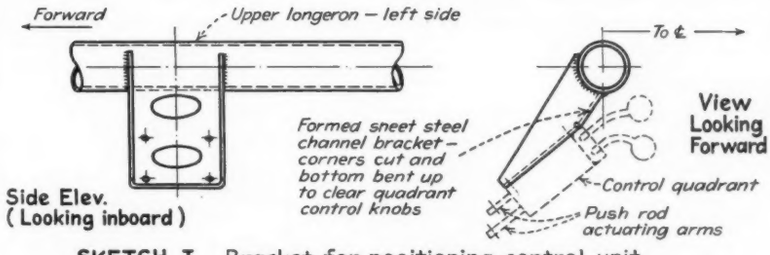
The plates used for this pulley bracket require two blanking and one forming die for fabrication, and the actual welding is complicated by the gap G being too narrow for easy welding, as it necessitates a weld overlap at the upper plate.

Fig. 31

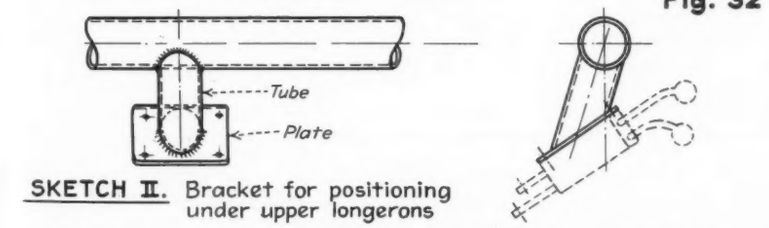
Cable-and-pulley type controls are used almost exclusively and the principal concern of the designer, after installing the control mechanism assemblies, is to provide suitable welded deflecting pulley assemblies within the fuselage tube structure for routing the cables to connect between control mechanisms and control surfaces. In Figs. 30 and 31 are shown examples of typical single and multiple pulley brackets. Pulley brackets must be designed so that there will be minimum distortion during welding. As the maximum permissible lead-off angle of a cable with reference to its pulley is 1 to 1½ deg., the design must be such that tooling can be made to positively locate the bracket in all three dimensions.

Another pulley bracket design consideration is the necessity of providing some type of cable guard to prevent the cable falling free of the pulley if it should become slack. In the designs shown here a rivet and rivet-spacer passing between the two plates of the pulley bracket serve as a guard. This guard should be located close to the pulley, and at the center of the cable arc in cases where this angle is small. In pulleys having a fairly large cable arc of contact between cable and pulley, use two guard pins, one where cable enters and another where cable leaves the pulley.

Practically every airplane is also equipped with certain auxiliary air controls for the purpose of trimming the airplane during flight—either in the form of an adjustable horizontal stabilizer, or as small movable surfaces known as tabs located in the trailing edges of ailerons, elevators, and/or rudders. The control systems for these usually consist of suitable handwheels in the cockpits, connected with cables to actuating unit assemblies in or near the movable surfaces.


Forward
Upper longeron — left side
To ℄
Formed sheet steel channel bracket — corners cut and bottom bent up to clear quadrant control knobs
View Looking Forward
Control quadrant
Push rod actuating arms
Side Elev. (Looking inboard)

SKETCH I. Bracket for positioning control unit outboard of longeron



SKETCH II. Bracket for positioning under upper longerons

Fig. 32

Obviously this bracket could not be used for the location shown in SKETCH I, since the scarf cut would run off the plate and would also require an extra long weld at the plate.
Conversely, the bracket shown at SKETCH I would probably need to be boxed in with a cover plate to develop sufficient torsional rigidity for locating the unit beneath the longeron.

Engine Controls

Engines must be provided with at least a throttle control, and in the case of high-horsepower military engines there may be as many as a dozen separate engine controls. Each separate engine control requires the installation of suitable supports for a bellcrank and push-pull rod system.

The majority of engine installations are designed with an engine control unit assembly which is placed in the pilot's cockpit. This is an assembly mounted on a panel and it includes the control levers for the engine throttle, carburetor mixture, propeller pitch, and sometimes a supercharger speed control. This unit is mounted to the left of the pilot, in a convenient position, and, as shown at Fig. 32, is usually bolted to a bracket welded to the upper left-side longeron.

(*Continued on next page*)

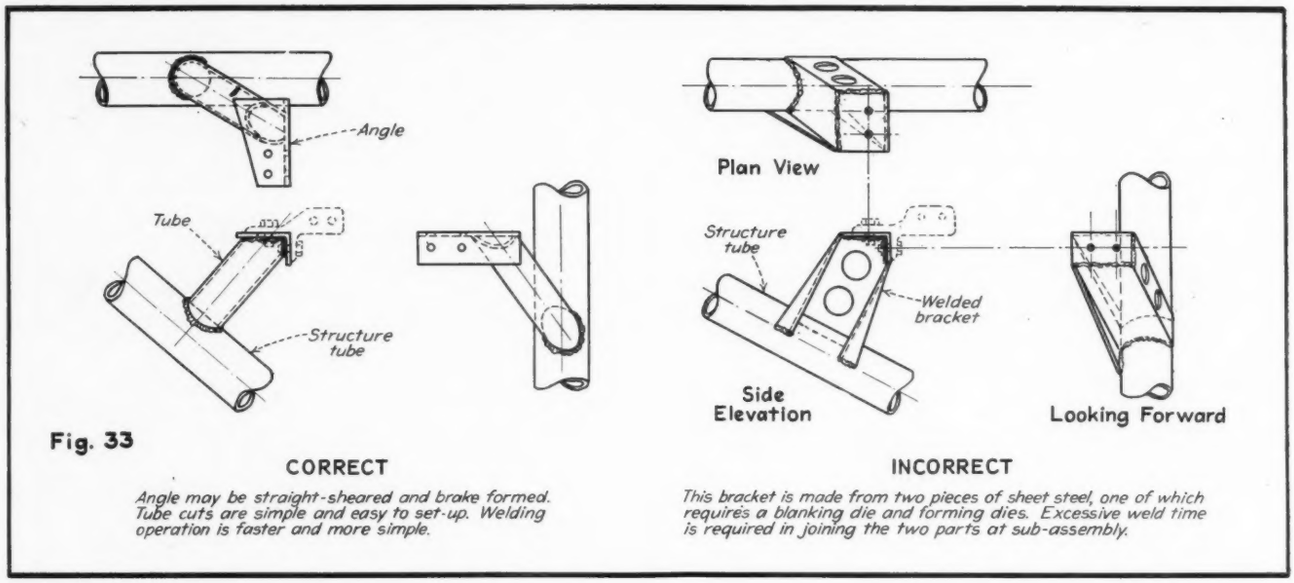


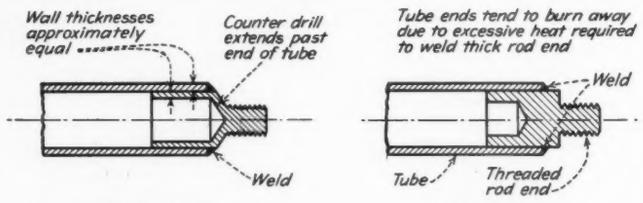
Fig. 33

CORRECT

Angle may be straight-sheared and brake formed. Tube cuts are simple and easy to set-up. Welding operation is faster and more simple.

INCORRECT

This bracket is made from two pieces of sheet steel, one of which requires a blanking die and forming dies. Excessive weld time is required in joining the two parts at sub-assembly.


Wall thicknesses approximately equal
Counter drill extends past end of tube
Weld
Tube ends tend to burn away due to excessive heat required to weld thick rod end
Weld
Tube
Threaded rod end

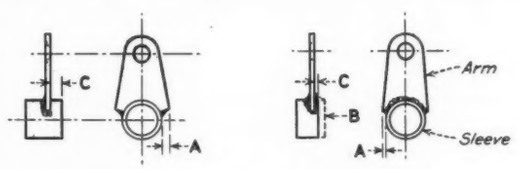
CORRECT

This type provides a more nearly balanced joint and heats quickly. The weld diameter may be held to approximately flush with the tube diameter. This is necessary when used through one piece fairleads. The shoulder has a slope to parallel the counterdrill angle and neutralize the effect of weld spill.

INCORRECT

Rod end walls are too thick, and counterdrill is not of sufficient depth. This joint would be easier to weld with the rod end telescoped to within about 1/16" or 1/8" of the shoulder, but the weld puddle would spill over and interfere with the check nut that is used with rod ends.

Fig. 34



CORRECT

C dimension is sufficient to permit welding, if desired, although but a single weld is shown. A dimension must be 3/16" minimum to prevent corners burning away during welding.

INCORRECT

Insufficient stock is allowed at A and corners may burn away during welding. Machining at B is undesirable as too much of the weld bead will be removed. Cannot be welded without machining, as C does not allow for normal weld fillet of 3/16".

Welding on both sides of an arm such as this is not incorrect if stress requirements necessitate such a design. In most cases, however, a single welded side will afford sufficient strength.

Fig. 35

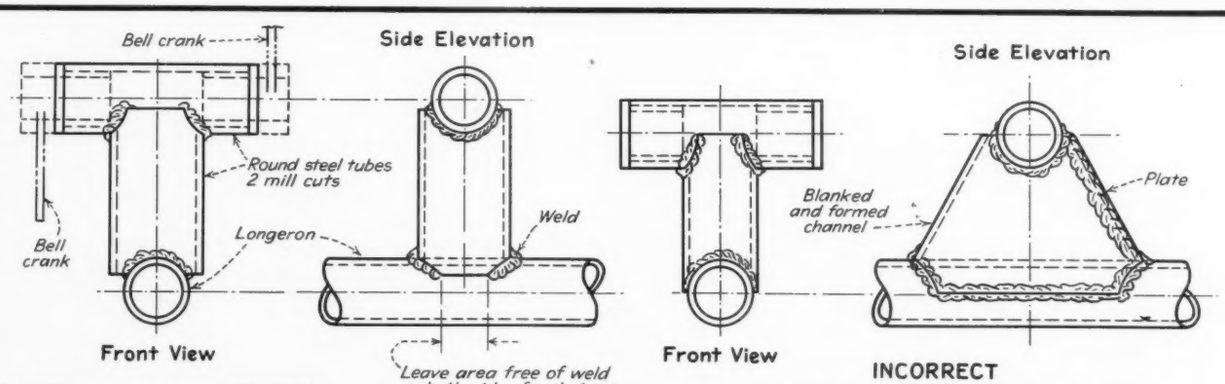


Fig. 36

CORRECT

INCORRECT

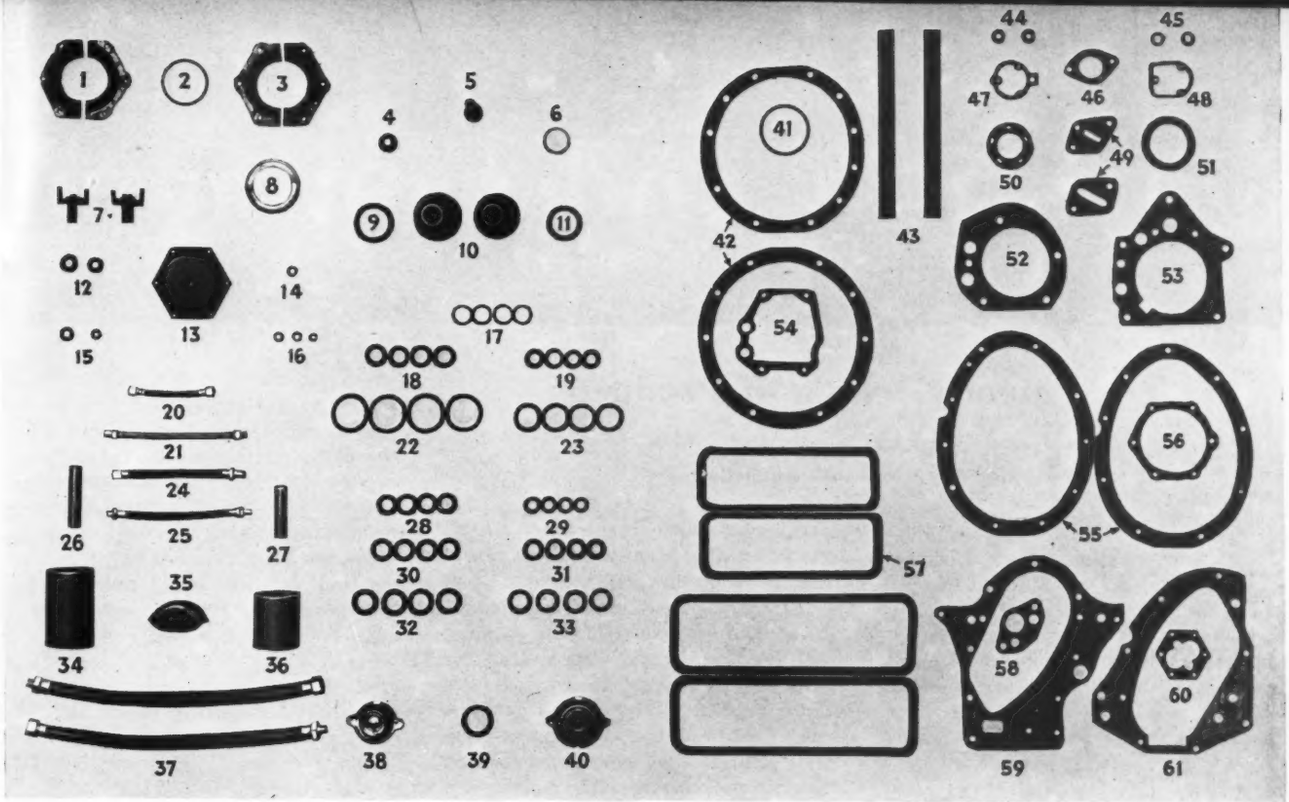
More weight due to extreme amount of weld necessary

Miscellaneous engine controls, such as for carburetor, air temperature, etc., are usually individual lever assemblies and, as shown in Fig. 33, can be attached to small brackets welded to the fuselage tube structure.

Typical examples of push-pull rods and bellcranks are shown in Figs. 34, 35 and 36. Push-pull rods are frequently made by welding threaded ends to a suitable length tubing, as shown at Fig. 34; using steel tubes for heavy loads, or by riveting the rod-ends to aluminum tubes for lightly loaded controls. Usually a ball bearing rod-end is screwed to each end of the rod and is pinned to lever or bellcrank with an axle bolt.

Welded bellcranks, for light loads only, have sleeve bearings, as in Fig. 35, and are used in pairs as shown in Fig. 36. Bellcranks for heavy loads are made of sheet metal and have ball or roller bearings.

(Part V will appear in February P.E.)



SYNTHETIC RUBBER

BECAUSE of its resistance to oils and its resistance to oxidation and aging effects, synthetic rubber is being widely adopted in the automotive industry, particularly for gaskets and for dust and oil seals. In some cars synthetic rubber hose in place of metal tubing is used in the fuel pump supply lines to deaden fuel pump noises. Below is a listing of some of the synthetic rubber parts used in cars built by Chrysler Corporation. In each application the type of synthetic rubber, Buna, Thiokol, Chemigum, or Neoprene, is chosen on the basis of specific physical and chemical requirements that must be met. Items marked (*) are coated or dipped.

N = Neoprene
B = Buna
C = Chemigum
T = Thiokol
LN = Ligno-Neoprene
NC = Neoprene-dipped cork
TP = Thiokol-coated paper

1 and 3. Rear main bearing seals (N)
2 and 8. Chain case cover seals (N)
4. Exhaust pipe bracket grommet (N)
5. Ignition coil nipple (N)
6. Steering gear tube oil seal (N)
7. Crankshaft rear bearing cap gaskets (N)
9 and 11. Spline shaft seal (propeller shaft) (B)
10. Propeller shaft boot (N)
12. Remote control gearshift bushing assembly (N)
*13. Underdrive transmisison vacuum cylinder diaphragm (C)
14. Transmission extension capscrew gasket (N)
*15. Underdrive transmission vacuum cylinder valve washers (N)
16. Transmission drive shaft retainer gasket (N)
17. Shock absorber—seal gasket (NC)
18 and 19. Shock absorber—seal packing (N)
20. Engine to oil gage hose (N)
21. Gas line hose assembly (T)
22. Shock absorber piston guide reservoir gasket (N)
23. Shock absorber—upper gasket (N)
24 and 25. Fuel line hose assembly (T)
26. Fuel tank filler vent hose (T)
27. Fuel tank vent alarm hose (T)
28. Front wheel upper control arm dust seal (N)
29. U-shackle seals (N)
30. Front wheel lower control arm bar bushing dust seal (N)
31. Front wheel upper control arm pivot bar dust seal (N)
32 and 33. Front wheel lower control arm pin dust seal (N)
34 and 36. Fuel tank filler tube hose (T)
35. Water pump by-pass elbow (N)
37. Fuel filter to injector hose assembly (N)
38 and 40. Radiator pressure cap gaskets (C)
39. Thermostat gasket (C)
*41. Oil pump cover gasket (LN)
42. Axle drive pinion carrier gaskets (LN)
*43. Oil pan end gasket (NC)
*44. Valve spring cover screw gasket (TP)
*45. Valve spring cover stud nut gasket (TP)
*46. Gasoline pump gasket (TP)
47. Underdrive transmission vacuum cylinder body gasket (TP)
48. Underdrive transmission vacuum cylinder solenoid gasket (TP)
*49. Fuel pump flange gasket (TP)
50. Fuel gage tank mounting gasket (TP)
*51. Chain case assembly oil seal gasket (TP)
*52. Transmission extension gasket (TP)
*53. Overdrive housing adapter gasket (TP)
54. Manual remote gearshift housing gasket (LN)
*55. Chain case cover gaskets (TP)
*56. Crankshaft rear bearing oil seal gasket (TP)
57. Valve spring cover gaskets (LN)
*58. Oil pump body gasket (TP)
*59. Gear case cover gasket (TP)
*60. Oil pump cover gasket (TP)
*61. Chain case cover plate gasket (TP)

Question and Comment

The Patent Marking of Products

HENRY CAVE
Consulting Engineer, Fuller Brush Company

In order to clarify the relative values of marking or not marking articles "Patent Applied For" or, when the patent is issued, with the patent number, I wish to present the legal phase of the matter to those concerned. This question of marking or not marking may be a serious consideration from the standpoint of tool and production factors, particularly when equipment such as molding dies is already in use.

There is no legal requirement whatever for such marking. In the case of the patent number, if such legal value as exists is to be obtained, the patent number should be put on so that it is clear and as reasonably visible as the maker's name.

The legal advantage is that damages cannot be collected by the patent owner if it is possible for the infringer to show to the court samples purchased on the open market without such marking. Of course, such an unmarked sample might have been produced prior to the issuance of the patent. The burden of proof would then be on the owner of the patent to prove that unmarked samples had not been distributed after the patent was issued with, of course, a reasonable period for providing means for marking the articles. The date of notification to defendant of the infringement would, therefore, become the earliest date from which damages could be collected. The decision of the court would merely constitute a cease and desist (from infringement) order.

The marking can be stamped on the finished article as a separate operation if this is found desirable, instead of being incorporated in the regular manufacturing operation. A small adhesive ticket could be applied giving this information, but it is improbable that this would be recognized by the court as allowing the collection of damages. Apart from damage collection, the patent marking is a friendly warning which may save trouble and expense to contemplated infringers both intentional and otherwise, and this could also be carried out to some extent by marking the patent number on all printed matter that would be issued relating to the article.

"Patent Applied For" is not required by the Patent Office and does not have any legal status. It is also merely a friendly warning to save everyone trouble either by being infringed or being penalized for infringement. As the Patent Office does not issue any information on what applications are filed in the office, unless they happen to come into interference, there can be no penalty for putting "Patent Applied For" without the actual application. It is, however, morally a misrepresentation. Legally, there does not seem to be any distinction between "Patent Applied For" and "Patent Pending" or its abbreviation "Pat. Pend.", although on analysis the latter statement might be construed to have more positive grounds than the former, such as office action on some claims, indicating that the patent would ultimately issue.

Pesqueira Supported on "Location of Jam Nuts"

To the Editor:

In your November 1940 number, under "Question and Comment," discussion is requested on the subject of "Location of Jam Nuts," presented by J. J. Pesqueira. In dealing with this subject there are two fundamental facts which should be kept in mind.

First, all bodies are elastic. That is, they deform on the application of an external force and, within their elastic limit, resume their former shape upon removal of the force. This is just as true of a block of cast iron as of a rubber band; it is only a matter of degree.

Second, all motion is relative, not absolute. We are here interested in preventing relative rotative motion between male and female threads.

Let us first consider two thick plates bolted together, and for the purpose of a theoretical analysis of their elastic effect, we may consider, as shown in Fig. 1, each plate as equivalent to two thin plates separated by springs. When the bolt is tightened, the springs compress and exert a force opposite to that applied. If the system were absolutely frictionless this force would cause rotative motion between the bolt and the nut and the bolt would unscrew. But the system is not frictionless and it is upon friction that we rely to hold the parts together. We will assume, however, that the nut is an easy running fit on the bolt and will ignore any friction caused by tightness.

The problem of relative motion will not be affected if we assume that either the bolt or the nut is fixed in space. For the sake of simplicity, we will assume that the bolt (together with the plates) is fixed. We then observe that the only thing available to prevent the nut from turning is the friction generated by the spring pressure between the underside of the nut and the plate, and between the male and female thread surfaces, marked "compression contact" in Fig. 1. The angle of the thread being less than the angle of repose, the system is in balance and the nut cannot turn, except under the influence of an external force.

Now, what kind of external force would be likely to loosen the nut? In all probability it will be vibration or shock transmitted through the plate. If this produces a continuous hammer blow on the underside of the nut, it might eventually produce a minute elongation of the bolt sufficient to release the pressure on the compression contacts. It is conceivable that under intermittent impact the friction might be momentarily so reduced that sliding would occur between the inclined planes of the male and female threads, allowing rotary motion. The question then is, what can we do to prevent this?

If we place a second nut, as shown in Fig. 2, screwing the first down tight and the second on the top of it, we shall probably get a better condition in practice owing to increased contact area, although the unit pressure would be reduced. In this case, it would appear that the inner nut would be right on the firing line and would be the first to take the shock. However, if vibration should cause either a permanent or instantaneous elongation of the bolt, the nuts

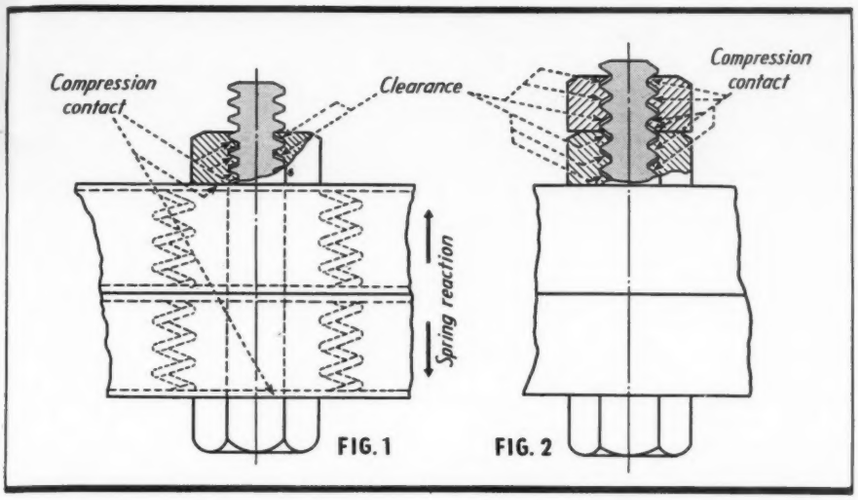


FIG. 1 FIG. 2

would still be jammed together and would react upon each other tending to maintain compression contact on the threads of both nuts.

The normal reaction of the compressed plates tends to break this contact in the inner nut, but the outer nut will still be firmly seated. Even if there should be a tendency for the first nut to turn, this could only occur at some instant when the nut is free and when the turning action would not be transferred to the outer nut.

It would appear that in this case, the outer nut is really the important one, though safe practice would indicate that each nut should be adequate to take the entire load. So, unless it is absolutely necessary, why make one nut thinner than the other? If it can't be helped, the outer nut should be the thicker one, as Mr. Pesqueira has stated. —JAMES O. G. GIBBONS
Bloomfield, N. J.

To the Editor:

My answer to your request for replies to J. J. Pesqueira's article is that he is correct in every particular. His fourth paragraph might be slightly amended by stating that the jam nut will carry the live load plus the load caused by the jam nut.

—FREDERICK FRANZ
New Haven, Conn.

Belt Tension Problem Assumes No Elasticity

To the Editor:

In the December number of PRODUCT ENGINEERING is presented E. W. Evans' solution of the rope tension problem offered in a previous issue. I have three comments to make on this:

The conditions prescribed require that motion be "imminent" or at a low constant velocity. For these conditions as well as for high speed operation, it is necessary to assume an inextensible belt as a premise to justify the analysis presented. The solution given, is the classical one offered in every elementary text in the field of engineering mechanics. It should be regarded as a first approximation, only. Where the modulus of elasticity of the belt material is low, the error in assuming the modulus to be infinite becomes prohibitive. If the belt stretches appreciably, within what might be called an elastic limit, an experimental verification of the results offered by Mr. Evans would be found impossible.

For high-speed operation, the theory presented takes no account of the effect of centrifugal force which always acts to reduce the normal pressure between belt and pulley. In addition to the stretch which operates to alter the character of the surface and therefore the coefficient of friction, there is also to be reckoned the effect on intimacy of contact brought about by a variable air entrainment. This influence is quite noticeable at high belt speeds.

Interested readers will find it worth while to consult "Belt Drives With Cast-Iron Pulleys and Paper Pulleys" in A.S.M.E. *Transactions of Machine Shop Practice*, for June 1932. This paper, by Norman and Moffat, presents experimental results obtained at Ohio State University. —GEORGE W. COLTON
Woodbridge, Conn.

Brush Manufacturer Comments on Nylon

To the Editor:

It is stated in the article "Nylon Filaments for Industrial Brushes" (PRODUCT ENGINEERING, Dec. 1940, p. 552) that stainless steel stem and tie wire for bottle brushes with nylon filaments has definite advantages over ordinary galvanized or brass wire. This statement, I believe, needs some further discussion.

Galvanized wire is the most generally used tie wire, but it does rust and discolor white nylon filaments. Also, tests in glass laboratories have shown that

Can You Work This One?

H. E. SMITH

This month's problem—

Brain Testers

1. Joe is 28 years old. He is twice as old as Jim was when he (Joe) was as old as Jim is now. How old is Jim?

2. A small motor weighing 15 lb. consists of rotor, frame, stator and bearings. Combined weight of rotor and frame is twice the weight of the frame and stator. The weight of the bearings is least of all. Assuming no fractions, what is the weight of the stator?

3. A man 5 ft. 5 in. tall is digging a hole. When asked how much deeper he is going he replies twice as deep, when the top of his head will be twice as far below the surface as it is now above the surface. How deep will the hole be?

4. Arrange the ten digits (0–9) to form numbers which will add to exactly 100, using each digit only once.

Solution to December problem—

Quickest Way Home

An engineer, in a rowboat 5 mi. perpendicularly off-shore from a point 20 mi. from his sea-side cottage, can row 3 mi. per hr. and can walk 5 mi. per hr. By rowing a distance Y diagonal to the shore and walking the distance X along the shore, he gets home in the quickest time, T. From a diagram,

$$Y = \sqrt{(20 - X)^2 + (5)^2}$$

$$T = \frac{X}{5} + \frac{\sqrt{(20 - X)^2 + (5)^2}}{3}$$

Taking the first derivative of T, reducing and setting it equal to zero to obtain a minimum value,

$$16X^2 - 280X + 75 = 0$$

Solving and substituting,

$X = 17.23$ mi., $Y = 5.72$ mi.

and the time was 5.34 hr.

it takes six times as much pressure to scratch glass with brass as it does with galvanized wire. For these reasons, brass is replacing galvanized wire. However, black nylon filaments, which cannot be discolored by galvanized wire, will be made available after the first of January—a fact which is expected to return galvanized wire to its popularity.

In my opinion, stainless steel wire has no great advantages in bottle brushes. It is harder even than galvanized wire, and may cause more severe scratching of the glass in the bottles. True, stainless steel is generally recognized as a sanitary metal, particularly in the food industry, but there is so much brass and galvanized iron used in equipment in breweries and soft-drink plants that engineers in these industries have come to accept these two cheaper metals as equally sanitary.

At the end of page 552 it is stated that white filaments are used in contrast to colored bottles so that broken-off bristles can be detected easily. This fact, although it sounds logical enough, has not been established to a certainty. From our present observation, the fact that black nylon filaments will soon be available may lead to the universal use of black filaments in bottle brushes.

May we state in passing that this is one of the fairest and most complete articles that we have ever had the pleasure of reading on nylon for industrial brushes.

—HARRY G. VOLCKENING
Volckening, Inc.

[*Editor's Note*—To replace the controversial second point in Mr. Volckening's discussion, we cite the small hand spotting brushes used in dry cleaning. These brushes are supplied with fine filaments for the delicate fabrics of women's clothes, and with coarser filaments for men's clothes. By coloring these white and black, respectively, the operator need waste no time selecting the proper brush.]

Micro-Midget Receivers Are Twice Actual Size

To the Editor:

The hearing aids illustrated as Fig. 5a and 5b of your article on "Permanent Magnets in Electrical Applications" (PRODUCT ENGINEERING, Dec., 1940, p. 561), are really not the "giants" as the article suggests. Actually they are real micro-midgets, not much larger in diameter than a 25-cent piece. The diagrams printed are really twice actual size.

Although this scaling up was necessary to bring out the details, it is important that engineers appreciate the minuteness of these receivers, otherwise the significance of the permanent magnet application will be lost. The tiny sintered magnets used in these hearing aids are probably the smallest ever commercially applied.

—DR. HERMANN SCHIEBLER
Sonotone Corporation

Case Histories in Patent Law

H. A. TOULMIN, JR.

NO. 20

QUESTION: When should a search be made to find out if an idea is new?

ANSWER: A search should be made to determine whether an idea is new when it is only an idea. The unfortunate mistake is usually made that the search to determine the novelty of the idea is delayed until after the major portion of the expense of developing the idea has been incurred. While the larger research laboratories make careful searches in the literature and the patent art before proceeding with any development work, yet in many of even the progressive organizations a great deal of work is done without any apparent check to see if the idea is new. This should never occur.

A large proportion of all the ideas of even experienced organizations turn out to be old. Indeed, such searches of the literature and prior patents may disclose practical reasons why a new idea is unworkable, which saves a great deal of experimental work and research.

Therefore, the first procedure should be to have a skilled searcher, who has knowledge of the art, search the Patent Office records to determine if this same idea has been patented in the United States or foreign countries. Coincident with that should be a careful search in the literature of that trade or art.

The way to bring these searches to a successful conclusion is as follows:

The Patent Office publishes a comprehensive loose-leaf book giving all the classes and sub-classes of patents as classified in the Patent Office by the classification division. This classification is also indexed according to the examining divisions. By way of explanation, each examining division has charge of certain classes. As an illustration, Division 50 has charge of chemistry, carbon compounds, coating, liquid coating compositions and plastic compositions, while Division 5 handles the classes of acoustics, buckles, buttons, clasps, harvesters, music, sound recording and reproducing and tying cords or strands. There is a further break-down in this interesting classification book by which all of the classes and sub-classes are alphabetically listed.

In addition to this vast storehouse of information about United States Patents, the scientific library of the Patent Office provides copies of patents as issued by the principal governments of the world. These patents are indexed and classified so that the expert searcher is able to find what he desires. One of the most important helps will be found in the official classifications of such foreign patents as are pertinent which are maintained by the examiners in each division. This is a good starting point for examining the more pertinent art and this collection of patents will give a lead into the proper classes to be examined. The Patent Office examiners are usually generous in their affording facilities to searching attorneys looking for anticipations in these prior art collections, and with few exceptions, afford them the facilities of their patent collections.

Another storehouse for valuable information is in the scientific library of the Patent Office where there are vast collections of catalogs, and trade magazines for both United States and foreign countries. These are not only indexed, but permission is granted to legitimate searchers to go into the vast "stack rooms" where the patents can be examined at first hand. In addition to these library facilities, the scientific libraries of the various departments afford valuable storehouses of information. The Bureau of Public Health, the Department of Agriculture, the Navy and War Departments, etc., all have collections of great value to the searcher.

In the automobile field, one of the greatest collections of prior publications, patents and literature is maintained by the National Automobile Association in New York. Among noted libraries where such searching is done is the John Crerar Library in Chicago, while some of the most valuable collections of specialized character are found in the files of the great universities.

News and Summaries

Protection Against Espionage and Sabotage Urged by F.B.I. for Industrial Concerns

Bureau offers "Suggestions for Protection of Industrial Facilities" to executives and cites cases of laxity in design engineering departments

THE FEDERAL BUREAU OF INVESTIGATION is now actively bolstering protection facilities of manufacturing establishments having large contracts to supply the government with defense material. No small part of this effort is being spent in establishing effective protection against possibilities of espionage and sabotage in design departments. For, as has been stated previously in these pages, the world at war fears the advanced designs now on the drawing boards of American industries or now in the preliminary stages of production. They fear the American system, which they can only match by copying. If developments of defense products can be kept secret, at least until mass production is started, it gives the owner of those developments a time advantage ranging from several months to a year or more over other countries. This applies not only to major defense weapons but also to materials and parts which are made by a large portion of American industry.

Accordingly, the Federal Bureau of Investigation directed the following letter to industrial executives through the trade press:

"In the war-torn world of today America must strengthen her internal and external fortifications if she is to survive the menace of totalitarianism. The protection of our industrial facilities to insure the uninterrupted production of goods and material is essential to our national defense. Based upon this premise the F.B.I. in September of 1939, in accordance with the request of the War and Navy Departments, inaugurated a program to survey the protective facilities of manufacturing establishments having large contracts to provide the government with defense materials. The purpose of the surveys is to submit recommendations to bolster the physical protective facilities of the manufacturing plants for the prevention of sabotage and espionage activities.

"The first and final responsibility to give speed and strength to our national defense program by protection against espionage and sabotage lies with industry itself. Only through the energy and alertness of its officials and workers will the full measure of preparedness be attained.

"In line with this plant survey program and to assist industrial concerns and municipalities in establishing effective protection against possibilities of acts of espionage and sabotage the F.B.I. prepared a comprehensive booklet entitled 'Suggestions for Protection of Industrial Facilities.' Because of its confidential nature this booklet is limited in distribution to heads of duly constituted law enforcement agencies and executive officials of industrial concerns manufacturing defense materials under government contracts.

"In response to the many questions as to how to obtain this publication I wish to state that a copy will be furnished an industrial concern upon the written request of an executive official."

—J. EDGAR HOOVER
Federal Bureau of Investigation

F.B.I. investigations have brought to light many examples of the laxity of engineering departments in providing adequate protection. For instance, a large safe with drawers for storing blueprints and development drawings was provided by one manufacturer in the design department. But the engineers did not use it. They felt it was perfectly safe to leave blueprints lying around, even at night. Although the personnel of the engineering department were trustworthy without a doubt, investigations disclosed that the men who entered the office at night to clean it up were not employees of the company, but were employed by a separate contractor cleaner. The manufacturer knew nothing of the record of the employees of the cleaning contractor, but inadvertently gave them convenient access to all design and production details. This dangerous situation was corrected by insisting that all drawings and data be locked in the safe when not in use.

In another case, the investigator was shown a safety vault located in the corner of the design department, which was on the first floor. This vault was provided with heavy locks, with steel flooring, roofing and walls. But, in the back of the vault was a window which was not even covered with a fly screen. Convenience of access through the window made the lock on the door look ridiculous. It was, the engineer explained, "necessary for light."

Propeller Vibrations Filmed in Test Flights

VIBRATIONS which occur in a propeller under flight conditions are determined in tests made at Wright Field by mounting automatic equipment consisting of a number of resistance pick-ups, batteries, amplifier, oscillograph, and collector rings, in the plane. Impulses recorded on the film in the oscillograph permit engineers to measure the vibration, determine its seriousness, and in many cases locate the source from the frequency and characteristics of the vibration lines on the film.

As engine power increases, vibratory tests become more important. With the development of a new experimental engine, it is necessary to design a propeller designed to absorb the full power of the engine. Complete vibration tests, including tests made under flight conditions when vibration may be caused by air gusts and interference of air flow due to the blade passing near the obstacles such as landing gears, fuselage, are necessary. Laboratory tests include suspending the propeller in an elastic sling and vibrating it under static conditions, propeller test rigs on which electric motors are used to whirl test new propellers, and test stands on which the experimental engine-propeller combination can be mounted and tested.

In flight testing equipment, resistance pick-ups, metal strips $\frac{7}{8}$ in. or more in length, are cemented on the propeller blades at the points where the stress is to be determined. Vibration in the propeller changes the linear measurement and the electrical resistance of the pick-ups. Current passes through

5,000 balls per hour are automatically checked for four dimensions, oversize, undersize, under-high limit and over-low limit, and sorted into five classifications by this gaging machine at Ford Motor Company. Balls for bearings are fed into a hopper at top, from which they roll to a gage head. If the ball is incorrect in any dimension, the gage operates relays to open one of four trap doors in a trough down which the ball rolls. Gage has micrometer adjustments for change-overs to different ball sizes.

---

these resistors from batteries, through the oscillograph and amplifier. Normal slight vibration which causes a fluctuation of electric current is recorded in a regular shallow wavy line. If abnormal vibration develops, the line becomes a jagged series of peaks which increases in size as the vibration increases.

Blast Furnace Gets Air-Conditioning Unit

ON HOT, HUMID DAYS, the air which is blown into No. 1 blast furnace at the Aliquippa Works of the Jones & Laughlin Steel Corporation may carry with it as much as 30 or 40 tons of water. This amount fluctuates with the moisture content of the atmosphere, and makes it necessary for the operator to guess the weather 24 hr. in advance and constantly change the proportions of the raw materials charged into the furnace, which approximate 2,000 tons of iron ore, 950 tons of coke, 500 tons of limestone, and 4,000 tons of air per day to produce 1,000 tons of molten pig iron. To eliminate this variation, air-conditioning equipment capable of removing from 7 to 40 tons of water per day will be installed. This equipment will maintain an air blast of constant humidity with which it will be possible to control the silicon content and iron temperature within narrow limits. Further advantages are seen in a possible reduction in the amount of coke and limestone necessary.

Color Television Demonstrated by G. E.

TWO-COLORED TELEVISION reproduction, in which the colors were claimed to be remarkably realistic, was recently demonstrated at Schenectady to members of the National Television Systems Committee by Dr. E. F. W. Alexanderson, consulting engineer of the General Electric Company.

A standard type receiver with no additional equipment other than a two-color 24-in. revolving disk about a foot in front of the picture end of the cathode ray of the standard type receiver was used. A similar colored disk revolved before the iconoscope pick-up tube of the transmitter. As each disk whirled at about 1,800 r.p.m., its transparent field of orange-red and greenish-blue reproduced images in faithful color without flicker. "In our early experiments we tried both two and three-color disks," Dr. Alexanderson said. "With two colors and a speed of 1,800 r.p.m. of the disk, the same color succeeded itself 30 times per sec. With three colors, they succeeded each other 20 times per sec., producing a color flicker. So that is why we decided upon the two colors for the present."

The demonstration is still in the development stage and General Electric has no plans for introducing color to its television programs for the present. Significance of the demonstration lies in the fact that the General Electric engineers feel that color television is practical with standard commercial receivers.

Do You Know That—

SUBSTITUTE FOR ASBESTOS in brake linings in Germany is aluminum or steel wire fiber. The filaments, 0.03 mm. in diameter, are bonded with synthetic rubber. (36)

AERATED UREA PLASTICS have been made experimentally. Possible uses are for heat and sound insulation, and as filler for aircraft wings to absorb gasoline leaks to avoid fire. (37)

MACHINE TOOLS in 1930 could remove only ¼ lb. of metal per min. By 1935, 17 lb. of chips per min. could be turned off—a figure which has since been considerably stepped up. (38)

EIGHT LB. OF URANIUM 235 would drive the Queen Mary across the Atlantic. Her bunkers now must carry 6,300 tons of fuel oil. (39)

ENGINEERING IN WASHINGTON

McGRAW-HILL WASHINGTON BUREAU, *PAUL WOOTON, CHIEF*

Washington engineering is British engineering and vice versa these days. For example, Dr. S. Zuckerman, of Oxford, reports that blast concussion injures men and animals by damaging their lungs. A heavy wrapping of heavy cloth or equivalent takes the punch out of concussion and saves the lungs. . . A special note tells how a center lathe is converted into a boring machine. A special gearhead replaced the chuck. . . The Ministry of Supply has appointed a "controller of plastics" to make sure that advantage is taken of this material.

Rapid obsolescence of war equipment is reported from Europe. Definite programs are planned by Germany and Great Britain to retire war equipment in step with design progress. If a type of plane seems likely to become obsolete in two years, German plans include immediate shifting of half the capacity on that type to the improved design. Some obsolescence is due to progress in other equipment, as when heavier tank armor renders 1-in. anti-tank guns ineffective. American policy accordingly expects early replacement of much equipment now being produced.

Insulation of X-ray with glass is claimed in a patent filed by Robert A. Miller of Tarentum, Pa. Advantage is that the operator of the machine can have protection, yet see the subject through the insulation. It is claimed the glass also intercepts ultra-violet light. The principle is high content of lead oxide; X-ray repellant characteristics of lead oxide are known. Such glass weathers easily and its insulation factor breaks down. Mr. Miller has put his lead oxide glass between two sheets of glass of high lime, silica and soda content, which protects the middle layer.

Wide angle aerial camera, Air Corps' latest development at Wright Field, carries 249 frames of film which, allowing for 60 percent overlap, are capable of taking 3,600 sq. mi. of area in one trip of the plane. Exposures are made at an altitude of about 20,000 ft. The plane carries a supply of oxygen in cylinders and the cabin is heated. Night photography from altitude is being tried with heavy flash bombs, which trip the shutter through a photoelectric cell. By slightly off-setting two exposures, the effect of a relief map is obtained, on which such minute things as wheel tracks can be seen.

Bad men with guns are always telling how quick they are on the draw. But talking a fast draw doesn't go, any more, because the FBI has rigged up a thing to measure shooting time. This is how it works—approximately. The agent stands with gun in holster, his gun hand on a spring switch. When he gets the order to fire he jerks his hand toward the gun and the switch starts the stop clock. When the bullet hits the target it breaks the circuit and stops the clock. The fastest men can draw and fire from hip or shoulder in a bit less than two seconds.

Perpetual motion machines, death rays, or even a fountain pen or a hot-and-cold shower-bath valve that would work—all are now welcome by the National Inventors' Council of the Department of Commerce. The Defense Commission too has a research committee that looks into all technical suggestions. Ideas are not only accepted, but are invited. Anything that will add to the efficiency, economy, or effectiveness of the defense program is worth submitting.

Bantam multi-purpose automobile for the Army, which the press office claims hasn't been stopped yet on cross-country tests—not even by 60 percent grades—is Washington's newest engineering job. The car weighs 1,350 lb., is 80 in. long, 40 in. high, and costs the government about $1,000 apiece.

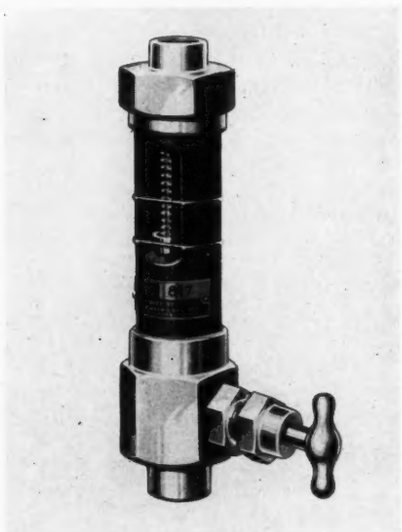
The midget is not intended for combat purposes. It has no armor; occupants are fully exposed. Primarily, it is a "command and reconnaissance" vehicle, which means scouting. It has many other practical applications. As a troop carrier it holds three men strapped in bucket seats. It could serve as a prime mover for the 37mm. anti-tank gun, which is mounted on a trailer; also could transport anti-aircraft guns.

Some specification and performance facts and figures: load, 500 lb.; 4-wheel drive enables it to crawl out of creeks, mud, brush; 4-cylinder, 45-hp., souped-up engine; speed 60–65 mi. per hr.; pull on the draw-bar, 1,000 lb. Puncture-proof tires have been sprayed with bullets and chopped with an axe—but kept on rolling.

New Materials and Parts

Pressure Sight Feed Oiler

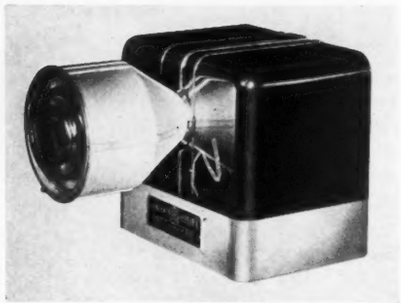
By installing this Fig. 817 pressure sight feed oiler at eye level in a convenient place, the operator can readily regulate the amount of oil delivered to out-of-the-way bearings. By noting the position of the indicating plunger, it tells the approximate amount flowing to the bearing. Body is of wrought steel



finished in black enamel, with polished nickel plated trimming. Needle valve permits adjustment from one-quarter pint to one quart per min., maximum working pressure of 50 lb. per sq. in. Available with ¼ or ⅜ in. tapped connections. S. F. Bowser & Co. Inc., Fort Wayne, Ind.

Photoelectric Relay

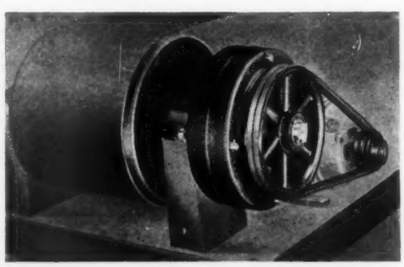
Attractively housed in a cast alloy chassis with a compact molded cover, this new relay will operate at speeds up to 150 interruptions per min. and at a minimum of 40 foot-candles at the photoelectric cell with not more than 15 foot-candles of extraneous light present. Applications include counting, sorting, weighing, measuring, controlling, signalling, inspecting and regulating when conditions permit. The relay has a controlled contact rating of 10 amp. at 115 volts a.c., with two normally open and two normally closed con-



tacts. The coil of the relay is energized when the light at the photoelectric cell is reduced below 30-foot candles. Relay can be mounted on wall or bench with rigid or flexible conduit, with open wiring, or by any other accepted method. General Electric Co., Schenectady, N. Y.

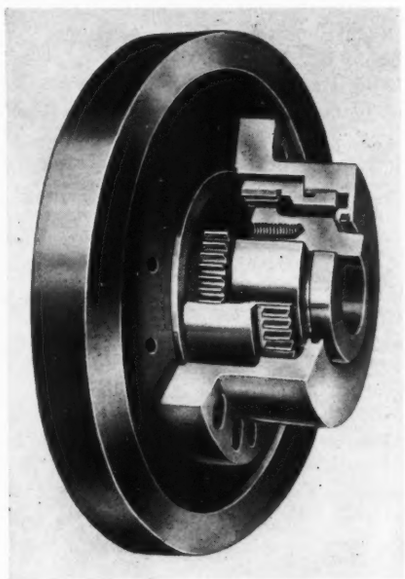
Speed Reduction Drive

This completely new and practical method of speed reduction consists of two major elements: a helical gear reduction unit which mounts directly on the shaft of the driven machine, and a standard belt drive between the motor and the input shaft of the reduction unit. The reduction unit has a standard fixed ratio of 13 to 1, any larger or smaller ratios being accomplished by proper selection of pulley sizes for the primary belt drive. The shock-absorbing action of the primary belt drive protects the gears from shock loads and the drive is claimed to have an over-all efficiency, when equipped with a tension-control motor base, approximating the 98½ per cent efficiency of the reduction unit itself. Five reduction units cover all applications from ½ to 30 hp., giving any desired speed between 11 and 250 r.p.m. American Pulley Co., Philadelphia, Pa.



Flexible Coupling

For connecting a shaft directly to a flywheel, brake drum or flange, this coupling provides complete flexibility with approximately one-half the axial clearance required by other types of couplings. The dummy, pilot or stub shaft generally necessary with other types of couplings is eliminated. Compensation for misalignment is made by an internal sleeve which floats between an externally geared hub and an internally geared covering sleeve. The internal sleeve which engages the hub and outer sleeve is free to slide and rock. Contact areas are large and



heavy oil lubricating the contact surfaces provides a cushion against shock without using flexible material. In connecting shafts of different diameters, the coupling is mounted on a smaller shaft and is bolted to a rigid, flanged, half coupling on the larger shaft. This permits the use of a smaller and lower priced coupling. Farrel-Birmingham Co., Inc., Ansonia, Conn.

Vibration Frequency Meter

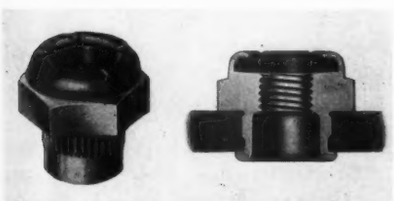
This new instrument is no larger than an engineer's slide rule, but it can indicate frequencies between 500 and 20,000

cycles per min. present in a vibrating body. This compact device is built around the principle of the vibrating reed and consists of a thin spring steel vibrator clamped at one end between a set of steel rollers. Knurled knob connected to the rollers moves the steel reed in or out, changing its frequency of vibration. A sliding pointer on the back end of the steel reed indicates the vibrating frequency. To use the meter,

its head is held against the vibrating body and the adjusting knob is rotated until the vibrator reed moves to and fro at maximum amplitude. If more than one vibrating frequency exists, there will be a point of maximum amplitude for each, and vibrations in differing planes may be detected by changing the axis of the meter. Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.

Self-Locking Nut

Self-Locking Nut with knurled shank is clinched permanently to sheet metal parts of assemblies which must frequently be taken apart. To install the nut, a hole is drilled in the structure and the shank is pressed into the hole. The mouth of the shank is then spread back against the structure to effect a



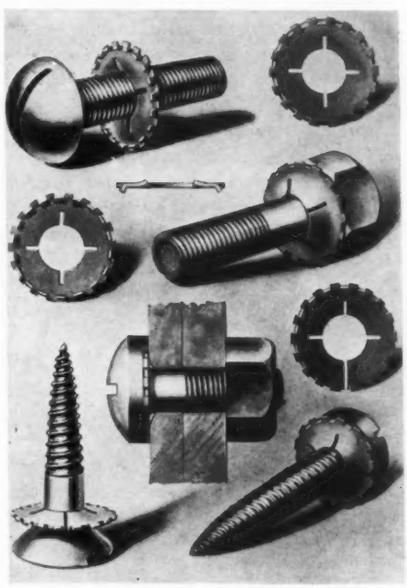
clinching hold. The knurling engages the drilled surface and prevents any turning of the nut. The head of the nut is fitted with the Elastic Stop Nut vulcanized fiber collar which resists the entrance of the screw, thus automatically taking up all thread play and bringing the load-carrying thread faces of nut and screw into a tight pressure-contact. Because of the resilient character of the fiber collar, the screw may be removed and replaced repeatedly without loss of locking action. Elastic Stop Nut Corp., 2332 Vauxhall Rd., Union, N. J.

High-Temperature Coating

Designed as a coating for metal parts which in application are subjected to temperatures of 400 to 500 deg. F., this material is claimed to hold its white color as well as its life at these high temperatures. This baking type material is applied by roller coating or spraying. For best results, a baking schedule of 325 deg. F. for approximately 20 min. is recommended. Watson-Standard Co., 225 Galveston St., Pittsburgh, Pa.

Pre-Fixed Lock Washer

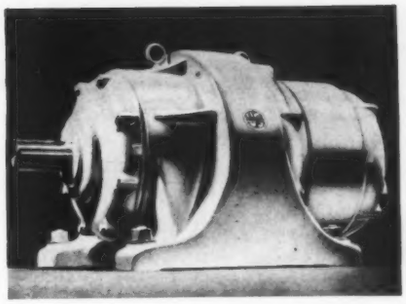
In contrast to previously known pre-assembly methods, this new lock washer is pre-assembled to standard threaded fasteners by a simple press operation.



The lock washer is a disk with a hole slightly smaller than the maximum diameter of the shank of the fastener. Slits extending radially from this hole form tongues which, when pressed over the fastener, prevent removal of the washer from the fastener during handling or in assembly. Mechanical Laboratories, Inc., 128-130 Latham St., Pittsburgh, Pa.

Gear Motors

Double and triple reduction geared motors up to 30 hp. at 91 r.p.m. incorporate the pyramidal gear pedestal de-



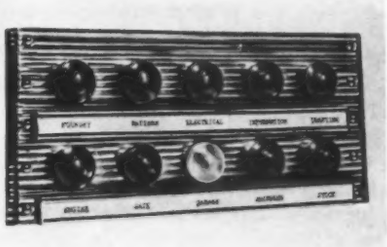
sign which provides ample support to withstand extra torsional strains and load shocks of geared power. All castings used in this new line of large Syncrogear motors are normalized, and permanent alignment of bearings and gears is assured. Both primary and secondary gears dip in a large oil reservoir in the pyramidal base, providing ample oil supply to all bearings and gears. U. S. Electrical Motors, Inc., Dept. 112, 80 34th St., Brooklyn, N. Y.

Rubberized Asphalt

A rubberized asphalt plastic material, which can be poured without heating into joints between concrete blocks, commonly known as expansion joints, adheres tightly to concrete or similar materials, and is permanently elastic. Small imprisoned air cells prevent extrusion of the material from expansion joints in hot weather. This improved elastic rubberized asphalt mastic is available in forms for building construction, expansion joints, in putty form for sealing glass to metal, for caulking wood seams, and in sheets or boards up to 8 ft. long and from ½ to 1½ in. thick. Special shapes are also available. M. G. E. Corp., 607 W. 25th St., Chicago, Ill.

Control Mounting Strip

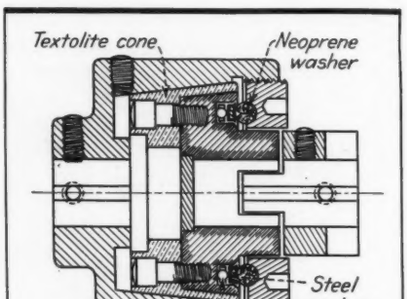
To facilitate the mounting in multiple of signal lights, toggle switches, push-buttons and similar devices, these standardized mounting strips are of stainless steel with a modern ripple finish and beveled ends. A strip is furnished with all devices mounted on it, ready to in-



stall. Strips are available in three lengths: for two units, for three units, or for five units per strip. One strip can mount any combination of devices, and several strips can be banked together. Each strip is mounted with two screws directly to machines, cabinets, panels, desk-tops, etc. H. R. Kirkland Co., Morristown, N. J.

Safety Clutch

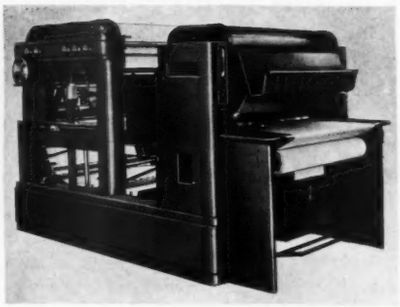
Flaton safety clutch can be set and locked at a predetermined load, and when load exceeds that point the clutch automatically slips to relieve undue strain on motors or breakage on driven shaft in applications such as conveyors and gear reductions. As shown on the diagram, the clutch consists of a housing into which is fitted a special friction cone made of Textolite, which has high heat-resisting qualities. This cone is backed up with a bronze thrust ball bearing which rides on a high-speed steel washer. Tension is maintained by a composition synthetic rubber pressure ring forced down by a take-up nut which can be locked at any desired working



torque. Flaton clutches are furnished in couplings, gears, pulleys, or sprockets, and are adaptable to machinery already in operation as well as to new equipment. Flaton Machine Works, St. Louis, Mo.

Blueprinting Machine

Model 22-16 blueprinting machine combines the exclusive features of the Model 22 printer with the Model 16 washing and drying machine. Primarily it is intended for engineering departments whose production requirements do not require the high speed of Model 22. Although the 24 ft. per min. actual printing speed of the Model 22 printer exceeds the 15 ft. per min. washing and drying speed of the Model 16 washer and dryer, only large prints can be fed at extreme speeds. Advantages are



sliding contact which smooths out all inequalities in tracings and gives 24¾ in. of uninterrupted exposure area; three-speed lamp control which allows the lamps to be operated on 10, 15 or 20 amp. as desired, and Actinic arc lamps which furnish uniform light emission, burn for 45 min. without interruption and resume instantaneously. C. F. Pease Co., 2601 W. Irving Park Rd., Chicago, Ill.

Brass Plating Process

Whereas previously heavy deposits of electroplated brass could not be obtained at all, this new brass plating process is two to four times as fast and makes possible smooth and bright electroplated deposits of 0.001 in. or more in thickness. Addition agents prevent chipping, improve luster, and prevent objectionable fumes. The relatively low temperature of operation, namely, 105 deg. to 125 deg. F., assures ease of control and economical operation. A uniform bright yellow color is obtainable over a wide range of current densities, hence uniform color over recessed objects is now possible. Known as the DuPont high-speed brass plating process, this new process will permit the specification of heavy thicknesses of plated brass. E. I. du Pont de Nemours & Co., Electroplating Div., Wilmington, Del.

Pillow Blocks

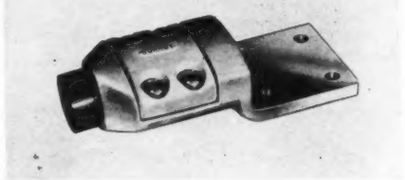
Labyrinth type seals with synthetic rubber seal rings are a featured improvement of the Ahlberg line of pillow blocks. These seal rings turn with the shaft and float in the housing, are frictionless and long wearing. Complete unit consists of full self-aligning, precision ball bearing, mounted in one-piece Parkerized accurately machined housing. The design lends itself to light and normal service where reliable yet inexpensive bearings are required. Ahlberg Bearing Co., 3062 W. 47th St., Chicago, Ill.

Porcelain Enamel

Pyroflex enamel, which burns at only 1,400 deg. F., permits the use of lighter gage enameling stock or black iron, and reduces burning temperatures approximately 200 deg. F. It is a low cost, genuine inorganic finish developed to compete with less durable finishes such as baked enamel. It will withstand heat and will not flake off when bent or flexed. Applied with one coat, no special metal bonding treatment is necessary. Pyroflex can be applied in a wide range of colors, effects and graining finishes, and requires only one-half the burning time of baked enamel. Porcelain Enamel & Mfg. Co., Eastern & Pemco Aves., Baltimore, Md.

Cable Terminal

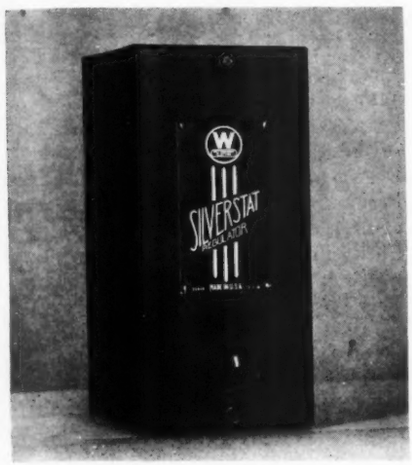
Designated as "Streamlined" cable terminal, Type SNS-H, the terminal is especially designed to facilitate easy taping of the connector and to help insure a moisture-proof seal to the insulation. A recess at the cable entrance



permits the end of the cable insulation to be inserted into the connector and thereby shrouds and protects the cut end of the insulation. The terminal cap, by means of hollow head capscrews, clamps the conductor as shown in the illustration. This type of connector is also made with hexagon head capscrews set in recesses in the cap in such a manner that the screws can be tightened with an ordinary socket wrench. Terminals are available for all cable sizes up to 3000MCM. Burndy Engineering Co., Inc., 459 East 133rd St., New York, N. Y.

Voltage Regulator

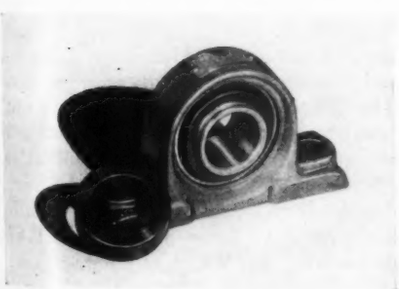
Designed especially for the automatic voltage control of small a.c. and d.c. generators, this new quick-acting rheostatic type Silverstat Junior regulator is available in d.c. ratings up to 25 kw. and a.c. ratings up to 44 kva. It quickly and automatically performs the same



function or operation as obtained by hand manipulation of the field rheostat of a machine under manual control. A change in voltage results in a corresponding change in the magnetizing effect of a coil, thereby causing an increase or decrease in attractive force causing the armature to seek a new position, thus either opening or closing in sequence an assembly of silver contact buttons. This shorts out or inserts steps of a carbon-plate resistor in series with the field circuit of the generator, keeping the generator output voltage within 3 per cent of normal. There are no parts requiring readjustment or replacement at frequent intervals. Westinghouse Electric & Mfg. Co., Dept. 7-N-20, East Pittsburgh, Pa.

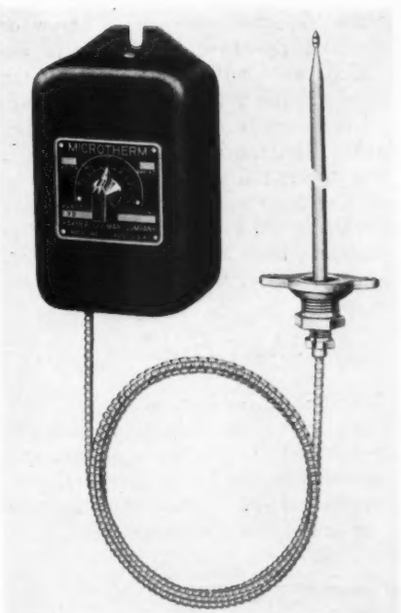
Sealed Transmission Units

New line of pillow blocks and other transmission units incorporating Fafnir "Mechani-Seal" ball bearings. These streamlined light series pillow blocks, flange cartridges and cylindrical cartridges offer the advantage of ease of application, made possible by the Fafnir Wide Inner Ring design, with a self-locking collar. They are locked to the shaft with a finger-twist, no machining, shaft shoulders, adapters or locknuts being required. In "Mechani-Seal" two steel plate shields form the innermost members and are tightly fitted to the bearing outer ring. An outer corrosion-proofed, steel plate shield, pressed on the inner ring, clears the inner plates by definite but close tolerances, and acts as an efficient slinger. The seal employs close tolerances rather than rubbing contact for its efficiency and imposes no friction or drag. The new transmission units are interchangeable with the separately sealed Fafnir units which preceded them, and are available in Types LAK (pillow block), LCJ (flange cartridge) and LC (cylindrical cartridge). Descriptive folder available. The Fafnir Bearing Co., New Britain, Conn.



Remote Bulb Thermostat

Suitable for all situations where a motor-operated valve or damper must be accurately and quickly positioned for control of such equipment as dryers, pasteurizers, food cookers, ovens, hot-



water tanks, and air conditioning. The remote bulb and bellows assembly are hydraulically formed providing uniformity and maximum strength. The volume of liquid and conducting surfaces are balanced to give minimum ambient temperature effect. Bulb is furnished with a mounting flange suitable for either immersion or duct mounting. Mechanism is inclosed in a black crinkle finish metal case with tight-fitting cover. Thermostats for 2-position control with snap acting contacts and for floating control are also available. Barber-Colman Co., Rockford, Ill.

Aircraft Bearings

Designed specifically for application where self-aligning bearings are required for control surface mountings and connections, landing gear controls, and other aircraft mechanisms, these self-contained single and double-row roller bearings are of large load capacity and maximum self-alignment ac-

tion. The Shafer roller bearing design is based upon the use of concave rollers running between convex raceways. Double-row self-contained aircraft bearings, having the rollers placed in an angular position with relation to the shaft, provide for thrust loads as well as radial loads. Single-row round end and eye bolt bearings are of the full roller type, providing high radial load capacity. Shafer Bearing Corp., 35 E. Wacker Drive, Chicago, Ill.

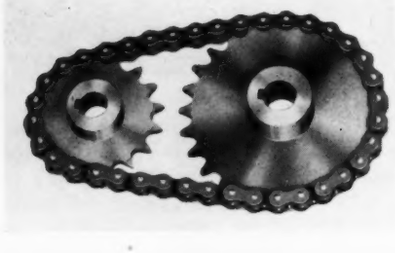
Metal Plating on Plastics

By forming a firmly adherent and highly conductive bonded coating on the surface of the plastic, metal platings are deposited on plastic surfaces by means of a process developed by the Metaplast Corp. Any metal that can be electroplated is suitable for this process. Parts of the surface can be masked to produce attractive designs and contrasts. Even extremely thin deposits of the plated metal can be buffed, are shock and wear resistant and withstand variations in temperature. This process makes possible the use of metal plated

molded plastic parts in place of solid metal parts. Portions of the plastic surface can be treated in order to provide for electrical conduction. Highly wear resistant deposits of chromium can be put on the plastic surface to increase wear resistance. Gears, rollers, valve parts are being made of laminated or molded plastic materials and plated, thus gaining the advantage of the lighter-weight plastic material. Metaplast Corp., 205-209 West 19th St., New York, N. Y.

8 mm. Roller Chain

In spite of its minute dimensions, 0.315 in. (8 mm.) pitch, with rollers 0.197 in. diameter by 0.125 in. width, No. 88 roller chain has 1,100 lb. average ultimate strength and is made to the same standards as larger Diamond chains. The material specifications, case-hardening and heat-treatment, are equally rigorous and exacting. Dimensional precision is identical. Each foot of chain is assembled to a total length tolerance of only +0.0156 in., and −0.000. This new chain is equally practical for dependable power transmission at high speeds and for applications for which maximum compactness



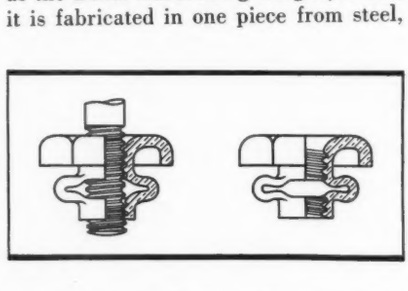
without sacrifice of enduring accuracy is essential, such as in positive driving of light weight machinery, for timing mechanisms, and precision instruments and equipment. The chain weighs only 1¾ oz. per ft. Diamond Chain & Mfg. Co., Indianapolis. Ind.

Alloy Steels

An extended line of special alloy steels especially for the aircraft industry now include heat-treated, cold-drawn annealed bar products. SAE steels of the various alloy series, such as 2300, X4130, 4340 and in the principal analyses required by the industry, are now supplied in bar form to meet the physical requirements for such parts as bolts, U-bolts, studs and other highly stressed parts. These aircraft steels are now available. The steels are melted in electric furnaces, rolled to close tolerances, and Magnaflux tested to meet the requirements of government specifications for the aircraft industry. Special care is exercised in their manufacture to insure uniform, clean steel of proper grain structure for use in aircraft parts. Jessop Steel Co., Washington, Pa.

Wing-Style Lock Nut

A vibration-resisting, self-locking nut of the "wing" style. When tightened the spring member of the nut expands to locking position, thus providing a constant tension. This nut presents all the advantages of the solid type nut, but is 10 to 50 per cent lighter and generally, size for size, costs less. Known as the Boots self-locking wing-style nut, it is fabricated in one piece from steel,



and cadmium plated. Sizes run from No. 6 through to ¼ in. Variations of this nut can be furnished for special purposes. In the illustration, at the left is shown the nut when not in use, at the right the nut is on a screw or bolt and the spring member is expanded to its self-locking position, the threads in the nut member pushing up, and the threads in the locking member down. The resulting constant tension locks the nut firmly in position. The nut has been specially developed for aircraft by the Boots Aircraft Nut Corporation. Distributed by Scovill Mfg. Co., Waterbury, Conn.

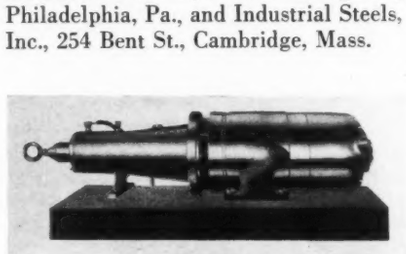
Sheet Holders

These sheet holders, which are readily applied with the forceps as shown in the illustration, provide a simple and convenient means for locating and temporarily holding metal sheets to each other and joining structural members.

It consists of a small steel cylinder with spring-actuated locking plunger. It can be inserted and removed from one side of sheet; access to both sides is not necessary. This Cleco sheet holder permits rivet hole locations and welding spot locations that would be impractical or not permissible by other known methods. Cleveland Pneumatic Tool Co., 3734 E. 78th St., Cleveland, Ohio.

Nitriding Stainless

Parts made of any stainless steel or stainless iron can now be Nitrided. This surface hardening process, known as the Drever-Industrial Stainless Nitriding Process, makes possible the formation of a tightly adhering glass hard (89-93 Rockwell 15N) abrasion resisting case which is from 0.002 to 0.028 in. thick. This Nitrided case has superior wear-resistant properties and practically the same corrosion resistance as the unnitrided surface. The surface can be ground or polished without loss of abrasion or loss of corrosion resistance. Parts of 18-8 stainless, Nitrided by the process have shown "B plus" rating in the standard A.S.T.M. salt spray test. Samples of both chrome iron and chrome nickel-iron, after Nitriding shows "A" rating after test in gasoline, kerosene and fuel oil. High pressure saturated and super-heated steam, impingement test on Nitrided 18-8 showed no deterioration of the case after 100 hr. The Drever Co., 746 E. Venango St., Philadelphia, Pa., and Industrial Steels, Inc., 254 Bent St., Cambridge, Mass.



Thrust Mechanism

Particularly adapted for operating levers, clutches, clamps, and similar equipment by remote push-button control, this new type of thrust mechanism directly translates revolutions into practical, straight line motion, delivering up to 4,000 lb. per sq.in. thrust. Thrust bar is direct-connected through differential gearing to a powerful torque motor. The Welco push and pull motor is not restricted to vertical or semi-vertical mounting, but can be used horizontally, upside-down or in any desired position. Reversible motor makes available a pulling motion of force equal to that of thrust. B. A. Wesche Electric Co., 1622–1628 Vine St., Cincinnati, Ohio.

Manufacturers' Publications

Materials

PRE-FINISHED METALS—American Nickeloid Co., Peru, Ill. Folder, 6 pages, 3⅝x8½ in. Briefly describes properties and advantages of pre-finished metals.

SHAPED WIRE—Page Steel & Wire Div., American Chain & Cable Co., Inc., Monessen, Pa. Bulletin DH-1226, 12 pages, 8½x11 in. Contains much useful information on grade, temper, and finishes. Illustrations of some of the many shapes available, tables and diagrams complete the data.

SMALL TUBING—Superior Tube Co., Norristown, Pa. Bulletin, 12 pages, 8½x11 in. Comprehensive discussion of fine small tubing, including method of manufacture and suggestions for applications.

STEEL—Crucible Steel Co. of America, 405 Lexington Ave., New York, N. Y. Bulletin AS 100, 48 pages, 4x8½ in. After introducing the subject of quality alloys steels, this data and stock list gives charts of mechanical properties and applications.

STEEL PLATES—Lukens Steel Co., 107 Lukens Bldg., Coatesville, Pa. Bulletin 107, 12 pages, 8½x11 in. Well illustrated booklet lists sizes of Lukens sheets and plates, and contains many tables of manufacturers' standard permissible variations for dimensions and workmanship.

Mechanical Parts

BEARINGS—Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill. Data Book 1775, 88 pages, 8½x11 in. Illustrates and describes "Friction Fighter Bearings." Tables of load ratings and minimum life expectancies, specifications, and engineering data facilitate selection of proper unit.

BEARINGS—Neveroil Bearing Co., Wakefield, Mass. Catalog 11, 15 pages 8½x11 in. includes descriptive and tabulated data on the 30 metal alloys and 12 wood formulas from which Neveroil bearings are made.

BELTINGS—B. F. Goodrich Co., Akron, Ohio. Catalog Section 2100, 4 pages, 8½x11 in. Sets forth features and illustrates "Highflex" brand of transmission belting.

CLINCHING NUTS—FabriSteel Products, Inc., Kerr Machinery Bldg., Detroit, Mich. Bulletin, 6 pages, 3⅜x8 in. Fast-On lock nuts for sheet metal parts are described in this bulletin, which includes a table of specifications.

CLUTCHES—Conway Clutch Co., 1543 Queen City Ave., Cincinnati, Ohio. Bulletin L-29, 8 pages, 8½x11 in. Gives complete description of Conway stud drive disk clutches, together with necessary dimensions, horsepower, torque data.

EXPANSION BOLTS—The Rawlplug Co. Inc., 98 Lafayette St., New York, N. Y. Wall chart, 15x20 in. Illustrated dimensional chart for expansion bolts and screw anchors gives many tables for wood and lag screws, machine bolts and machine screws.

FLEXIBLE HOSE—Pennsylvania Flexible Metallic Tubing Co., 72nd St. and Powers Lane, Philadelphia, Pa. Bulletin 52-G, 8 pages, 8½x11 in. Illustrates and describes Penflex bronze steam hose and clincher couplings. Bulletin 59-G, 8 pages, 8½x11 in. Gives engineering data for galvanized steel hose and couplings.

LEATHER PRODUCTS—Charles A. Schieren Co., 30-38 Ferry St., New York, N. Y. Bulletin 1962, 24 pages, 8½x11 in. Covers flat and round leather belting, leather packings, and leather specialties and accessories. Horsepower ratings for oak-tanned leather belting are included.

PIPE FITTINGS—Taylor Forge & Pipe Works, P. O. Box 485, Chicago, Ill. Catalog 401, 216 pages, 8½x11 in. Includes much technical data on design, pressure and temperature ratings, flange bolting, new test analyses, accepted standards, etc., for complete line of seamless steel pipe fittings.

PUMPS—Allis-Chalmers Mfg. Co., Milwaukee, Wis. Bulletin B-6140, 6 pages, 8½x11 in. Describes and illustrates the features of the new all-in-one "Electrifugal" pump for 1 to 10 hp. for heads up to 106 ft.

REFRIGERATOR COMPRESSORS—Worthington Pump & Machinery Corp., Carbondale Div., Harrison, N. J. Bulletin C-1100-B11, 6 pages, 8½x11 in. This bulletin gives a complete pictorial description of the vertical two-cylinder type Worthington-Carbondale refrigeration compressors.

Electrical Parts

CONTROLLERS—American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport, Conn. Catalog, 84 pages, 8½x11 in. Covers complete line of American self-operated controllers, combined temperature controllers with indicators, pressure controllers, wide-range indicating controllers, wide-range non-indicating controllers and controlographs.

MOTORS—General Electric Co., Schenectady, N. Y. Bulletin GEA-1542D, 4 pages, 8x10½ in. Illustrates and describes features of heavy-duty d.c. motors, constant speed from 1 to 60 hp., and adjustable speed from ½ to 15 hp. Bulletin GEA-1868B, 4 pages, 8x10½ in. covers heavy duty d.c. motors, constant speed from 50 to 300 hp. and adjustable speeds from 15 to 200 hp.

MOTORS—Reliance Electric & Engineering Co., Ivanhoe Rd., Cleveland, Ohio. Bulletin 119, 4 pages, 8½x11 in. Well-illustrated bulletin sets forth features of Type AA Reliance induction motors, with ratings from 25 to 250 hp. at 1,200 r.p.m., 2 and 3-phase.

MOTORS—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Leaflet F-8498, 2 pages, 8½x11 in. Universal governor-controlled motors, available in sizes from 1/50 to 1/20 hp., for a wide range of quiet, constant speed applications, are described.

MOTOR CONTROLS—Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee, Wis. Catalog CS-395, 36 pages, 8½x11 in. Comprehensive discussion of Unitrol, the recently announced method of centralized motor control.

MOTOR STARTER—General Electric Co., Schenectady, N. Y. Bulletin GEA-2064B, 6 pages, 8x10½ in. This bulletin, in picture-caption style, describes the new D-51 magnetic motor starter.

PHOTOELECTRIC CONTROL—Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa. Price List 18-312, 8 pages, 8½x11 in. Illustrates common applications of the Photo-Troller, and includes handy application charts showing what unit to use.

Miscellaneous

INSTRUMENTS — General Electric Co., Schenectady, N. Y. Bulletin GEA-2753, 4 pages, 8x10½ in. Brief description of nine General Electric instruments and time switches for general industrial use.

OPTICAL STRAIN GAGE—American Instrument Co., 8010-8020 Georgia Ave., Silver Spring, Md. Bulletin 2084, 8 pages, 8½x11 in. Contains comprehensive information describing the Tuckerman optical strain gage for measuring tension and compression strains as small as 0.000002 in.

THERMOMETERS AND PRESSURE GAGES—Brown Instrument Co., Wayne & Roberts Aves., Philadelphia, Pa. Catalog 6706, 20 pages, 8x10½ in. Covers complete line of vapor-actuated, gas-actuated, mercury-actuated and portable thermometers for indicating, recording and controlling processing operations.

VOLTMETER—General Electric Co., Schenectady, N. Y. Bulletin GEA-3020A, 4 pages, 8x10½ in. General description including price listing and illustrations of portable electrostatic voltmeter.

Books and Bulletins

Graphical Treatment of Vibration And Aircraft Engine Dampers

C. H. POWELL—*288 pages, 5½ x 8¼ in., 224 curves, charts and diagrams. Blue clothboard covers. Published by Bookcraft, Inc., 135 Johnson St., Brooklyn, N. Y. Price $7.50.*

In this text primarily of interest to automotive and aircraft engine designers, Dr. Powell has developed graphical systems for the treatment of vibration problems. Graphical systems are built up from the symbolic reduction of an oscillating motion to a straight line. Certain elemental mechanical forces, elastic, frictional or inertial, are considered under the action of an oscillatory torque. Also the interaction of two oscillating members coupled elastically and frictionally are studied. The solutions of these problems subjected to graphical synthesis, are built up into frameworks by the process of adding them together vectorially. Once the framework or vector diagram is constructed, amplitudes and phase differences of the vibrating members are derived through application of the properties of right-angle and similar triangles. The method is not intended to replace analytical deductions, but rather to supplement them.

No catalog descriptions, photographs or drawings of dampers are included because of the constant change in damper design. Tradenames and inventors' names have also been avoided. All charts and graphs were drawn by the author, most of the charts being presented in large scale on inserted folded sheets for easy reference and use.

Materials Handbook

GEORGE S. BRADY—*Fourth Edition, 591 pages, 6½ x 9¼ in. Brown imitation leather covers. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price $5.*

Revisions made necessary by developments in and additions to the ever-growing list of engineering materials since the third edition was published in 1937 have been included by the author in this new edition. Style of presentation remains substantially unchanged. General information, with the most commonly used comparative figures, is given on materials in their group classifications in order to give a general picture patented and trade-named materials are then described to give a more specific understanding of commercial applications. Classifications are listed alphabetically and a cross-index facilitates reference.

A Handbook of English in Engineering Usage

A. C. HOWELL—*402 pages, 5x7¼ in. Clothboard covers. Published by John Wiley & Sons, Inc., 440 Fourth Ave., New York, N. Y. Price $2.50.*

Reports, specifications, society papers and articles written by engineers are usually perfect in grammar and spelling but only too often glaringly deficient in all other respects. Professor Howell evidently recognizes this because he devotes a goodly portion of his excellent book to explanations of the requirements for writing that will be easy to read. Thus, the first chapter, "Characteristics of Engineering Writing" discusses the fundamentals of good writing which include clarity, conciseness, completeness, accuracy and unity. Each of the fundamentals is then analyzed in detail.

It is difficult to do this book full justice in a brief announcement. The author has covered thoroughly the writing of articles, reports, specifications and letters. Not only has he made a remarkably simple and clear analysis of good writing, but he has also discussed the general requirements for good mechanical appearance and the use of charts and illustrations. This book is not merely a text on good grammar and the rules of writing; it is a clear and simple exposition of the fundamental requirements for good writing. Professor Howell is to be congratulated on having written an excellent book. Every engineer will profit by reading it, studying it and using it for reference.

Writing the Technical Report

J. RALEIGH NELSON—*373 pages, 6¼ x9¼ in. Green clothboard covers. Published by McGraw-Hill Book Co., 330 W. 42nd St., New York, N. Y. Price $2.50.*

Preparing a technical report is just like the job of designing and constructing an engineering structure or machine. Upon this fundamental Mr. Nelson bases his purpose in writing his book, which proceeds swiftly and directly to develop the principles of report writing, to illustrate them and show how they may be applied to the procedures involved in the design of a report. The author aims at fundamentals—fundamentals that are necessary when none of the popular and convenient blank forms can serve.

The book consists of four parts. The first presents a review of fundamental considerations. The second gives specific directions for the set-up of the report, with many examples. The third outlines a systematic procedure for the critical examination of a report, and the last part suggests a series of assignments for study.

Mr. Nelson's clear, free-running style makes the book extremely easy to read and proves his authority on the subject of exposition. Breakdown by subject, the copious use of examples, and numerous suggestions make it easy for the reader to attack and improve his individual shortcomings.

• • •

Beetle Molding Materials

46 pages, 8½x11 in. Published by American Cyanamid Co., Beetle Products Div., 30 Rockefeller Plaza, New York, N. Y.

The plastics industry is not static, and every once in a while a booklet like this fills a great need for up-to-date technical data. Design of molded parts, molding procedures, and properties of urea plastics sold under the tradename Beetle are thoroughly discussed, with tabulated data, design sketches, and photographic illustrations included wherever helpful. The design section is particularly good; sketches show how sections should be designed for even flow and cure, where allowances should be made for shrinkage, the design of inserts, and points where cracking may occur on aging.

Analysis of Changes in Transportation

Vol. I, 107 pages, 8½x11 in. Published by Edward G. Budd Manufacturing Co., Philadelphia, Pa.

Light-weight, streamlined passenger trains, made possible by new stainless steel welding methods and diesel engines, have revolutionized and revivified railway passenger travel. The sweeping trend is traced pictorially in this easy-to-read Budd analysis, which begins its story at the decline of the railroads with the advent of the automobile and truck, presents a cross-sectional view of railroad traffic at the time when the first streamlined Zephyr was introduced and shows the impact of the rapidly growing number of light-weight units on passenger travel.

BALL, ROLLER AND NEEDLE BEARINGS

Relative Capacities and Characteristics

The following table compares inherent characteristics of different types of rolling contact bearings of different design. Load capacities tabulated are the average for bearings ranging from ½ in. to 4 in. bore diameter, and are determined in the following manner:

(a) Load ratings as given in manufacturers' catalogs were reduced to equivalent rating of 300 r.p.m. for a minimum life expectancy of 3,000 hr. by using the formulas presented in the article, "Ball and Roller Bearings" appearing on page 12 of this number.

(b) The projected area of the bearing was determined as indicated in the diagram at right.

(c) The reduced rating was then divided by the projected area to obtain load capacity in lb. per sq.in.

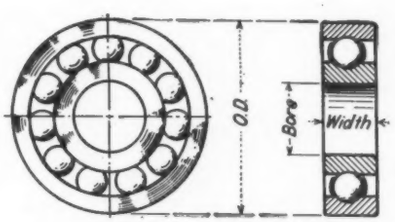
The factor *X* in the load-capacity column is the multiplying factor for converting from the load capacity given, axial or radial, to the one not given in the table, according to the formula:

Radial Load = *X* × *Axial Load.*

Finish of bearings is classed as regular (reg.), precision (prec.), and ultra precision (u. prec.). Shock load capacity is tabulated as the factor by which the load capacity is to be multiplied to give the capacity under shock conditions specified.

Relative magnitude of bearing friction is expressed numerically by rating them from 1, least friction, to 5, greatest friction. But these numbers are not factors, nor do they infer that a bearing with a friction index of 5 has five times the friction of an index of one.

Noise of bearings is rated 1, 2 or 3, the index 1 indicating least noisy, and 3 the most noisy.



The cross-hatched area is the projected area of the ball, roller or needle bearing. For bearings wherein the balls, rollers or needles operate directly on the shaft, the shaft diameter is used in place of the bore

Ball Bearings

Bearing Type	Load Capacity: Steady Load lb. per sq. in. projected area: Duty	Radial	X	Axial	Shock Load: Degree 1=steady 3=shock	Load Factor	Finish	Maximum Speed hundreds of r.p.m.: Dia. of Bore or Shaft, in.: ½	1	2	4	Relative Friction 1=Min. 5=Max.	Relative Noise 1=Min. 3=Max.	Degrees Misalignment Allowed	Minimum No. of Bearings to Stabilize Axially
BALL BEARINGS — OPEN TYPE															
Open Type.....	..	1,050	1 to 1.2	....	1 2	1.00 0.33	Prec. U. Prec.	250 500	150 320	.. ..	.. ..	1 1	1 1	0 0	2 2
BALL BEARINGS — SINGLE WIDTH															
Deep Groove.....	L	1,075	0.8	....	1	1.00	Reg.	75	60	45	20	2	2	±1	1
	M	975	to	....	2	0.50	Prec.	200	150	95	45	1	2	±1	1
	H	875	1.5	....	3	0.33	U. Prec.	450	300	..	..	1	1	±1	1
Filling Slots.....	L	1,500	2.0	....	1	1.00	Reg.	75	60	45	20	2	2	±1	1
	M	1,250	2.0	....	2	0.50	Prec.	200	150	95	45	1	2	±1	1
	H	1,150	2.0	....	3	0.33									
Angular.....	L	....	0.50	3,200	1	1.00	Reg.	75	60	45	20	2	2	0	2
	M	....	to	3,000	2	0.50	Prec.	200	150	95	45	2	2	0	2
	H	....	0.33	2,900	3	0.33									
Spherical..... (self-aligning)	L	975	2.5	....	1	1.00	Prec.	200	150	95	45	1–2	2	±5	1
	M	825	to	....	2	0.50									
	H	750	4.5	....	3	0.33									
BALL BEARINGS — DOUBLE WIDTH															
Deep Groove.....	L	1,050	1.1	....	1	1.00	Reg.	65	50	30	15	2	2	0	1
	M	950	1.1	....	2	0.50	Prec.	150	100	80	35	2	2	0	1
	H	825	1.1	....	3	0.33									
Angular.....	L	1,475	0.833	....	1	1.00	Reg.	55	45	35	15	3	2	0	1
	M	1,275	to	....	2	0.50	Prec.	140	95	70	30	2	2	0	1
	H	1,075	1.2	....	3	0.33									
Spherical..... (self-aligning)	L	825	1.5 to	....	1	1.00	Prec.	200	150	95	45	2	2	±5	1
	M	725	3.0	....	2	0.50									
					3	0.33									
BALL BEARINGS — AXIAL															
Groove Type															
½–2 in. bore..	..	0	—	1,800	1	1.00	Prec.	50	35	12	5	4–5	3	0	2
1¾–4 in. bore..	..	0	—	2,200	2	0.50									
					3	0.33									

Roller and Needle Bearings

Bearing Type	Load Capacity: Steady Load lb. per sq. in. projected area: Radial	X	Axial	Shock Load: Degree	Load Factor	Finish	Maximum Speed hundreds of r.p.m.: Dia. of Bore or Shaft, in.: ½	1	2	4	Relative Friction 1=Min. 5=Max.	Relative Noise 1=Min. 3=Max.	Degrees Misalignment Allowed	Minimum No. of Bearings to Stabilize Axially
ROLLER BEARINGS — SINGLE WIDTH														
Cylindrical Rollers														
With inner race	1,650	—	0	1	1.00	Reg.	..	50	30	18	2	2	0	*
Direct on shaft	2,225	—	0	2	0.77	Prec.	..	150	95	45	1	1	0	*
Cylindrical Rollers														
Races with lips	1,500	..	†	1	1.00	Prec.	..	75	45	25	2	2	0	1–2
				2	0.77									
				3	0.50									
Cylndrical Rollers														
Full type	2,000	—	0	1	1.00	Prec.	..	20	15	8	4–5	3	0	*
(no retainer)				2	0.77									
				3	0.50									
Tapered Rollers-														
Retainer type	....	....	....	1	1.00	Reg.	..	50	40	18	4–5	2–3	0	2
Reg. angle	1,500	1.43	....	2	0.77									
Steep angle		0.70	1,450	3	0.33									
Full type														
Reg. angle	1,775	1.39	....											
Steep angle	....	0.73	1,700											
Hourglass Rollers														
Spherical	....	....	....	1	1.00	Reg.	..	50	40	18	4–5	2–3	±5	2
⅝–1¾ in.	1,100	0.9	....	2	0.77									
1½–4 in.	1,750	1.0	....	3	0.33									
ROLLER BEARINGS — DOUBLE WIDTH														
Cylindrical Rollers														
With inner race	1,725	—	0	1	1.00	Reg.	..	45	28	16	3	2	0	*
Direct on shaft	2,325	—	0	2	0.77									
				3	0.50									
Wound Rollers														
With inner race	525	—	0	1	1.00	Reg.	..	20	20	15	4–5	3	0	*
Direct on shaft	675	—	0	2	0.77									
				3	0.35									
Tapered Rollers														
Double bearing				1	1.00	Reg.	..	50	40	18	4–5	2–3	0	1
Up to 3 in.	1,200	2.75	..	2	0.77									
Over 3 in.	1,750	2.75	..	3	0.33									
Barrel Rollers														
Spherical				1	1.00									
Light series	2,150	3	..	2	0.77	Prec.	..	..	..	30	4–5	2	±5	1
Medium series	1,850	2	..	3	0.33	Prec.	..	..	75	30	4–5	2	±5	1
Hourglass Rollers														
Spherical	1,900	2	..	1	1.00	Reg.	..	..	40	18	4–5	2–3	±5	1
1⅛–4 in. bore				2	0.77									
				3	0.33									
ROLLER BEARINGS — EXTREME WIDTH														
Wound Rollers														
With inner race	750	—	0	1	1.00	Reg.	..	20	20	15	5	3	0	*
Direct on shaft	925	—	0	2	0.77									
				3	0.35									
ROLLER BEARINGS — AXIAL														
Cylindrical Rollers														
2–5 in. bore	0	—	3,000	1	1.00	Prec.	..	..	15	10	5	3	0	2
				2	0.77									
				3	0.50									
Conical Rollers														
3–5 in. bore	0	—	3,500	1	1.00	Reg.	..	..	..	8	5	3	0	2
				2	0.77									
				3	0.50									
NEEDLE BEARINGS														
With Inner Race				1	1.00	Prec.	..	50	40	18	4–5	2–3	0	*
¾–1½ in.	2,500	—	0	2	0.68									
1⅝–4 in.	4,300	—	0	3	0.34									
Direct on Shaft				1	1.00	Prec.	..	50	40	18	4–5	2–3	0	*
¾–1½ in.	3,900	—	0	2	0.68									
1⅝–4 in.	6,900	—	0	3	0.34									

* Additional thrust bearing required. † For axial location only.